AUTOMOBILE ENGINEER

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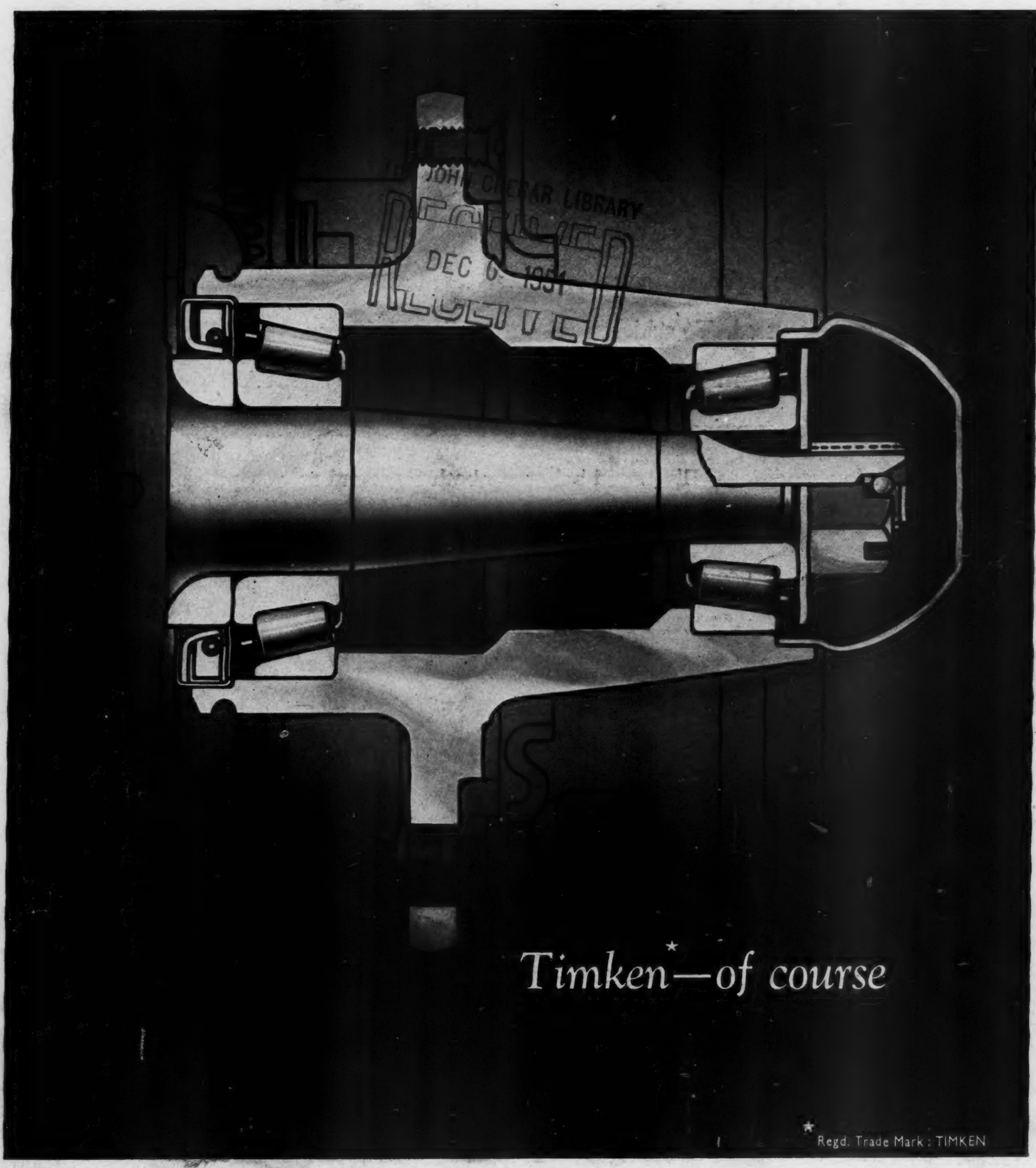

AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

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Gear Life

IN recent years shotpeening to increase fatigue resistance has been applied successfully to several types of components, both in this country and in America. English practice in the main has been confined to work on springs, but in the U.S.A. the process has also been fairly widely used on gear teeth. Interesting information concerning the effect that shotpeening the teeth can have on gear life was given in a paper presented by J. C. Straub to the Spring Meeting of The American Society of Mechanical Engineers, on which these notes are based.

There are three forms of gear failure: (a) through breakage of the teeth; (b) through pitting, usually on the pinion at or somewhat below the pitch line; and (c) through scoring. Breakage and pitting are generally considered to be fatigue effects caused by bending and compressive stresses respectively. Scoring, or roughening of the tooth flanks, is probably due to the welding and subsequent tearing apart of working surfaces that slide upon each other at high velocity and under high compression. It is likely to occur at early stages in gear life, if at all.

Fatigue Failures

Since so many variables enter into the question of failure through breakage or pitting, it was thought that the most reliable data would be obtained by statistical analysis of a large number of test results. Therefore complete information on the design factors and dynamometer fatigue test results for spur and helical automobile type transmission gears was obtained from several different manufacturers. Although this showed that parts which are apparently identical are subject to variation in fatigue, it is possible to obtain average values. Stresses computed by several different methods were then plotted logarithmically against the average values of fatigue life in cycles. The method of computation that gave the most consistent relationship between the calculated stress and the fatigue life was then selected. A value thus obtained is not necessarily accurate, but a good estimate of fatigue life may be made from it.

The method finally chosen assumed that the load on the teeth was distributed uniformly on the average total length of contact lines, and the tooth strength factor was obtained in a manner described by Wilfred Lewis in *Automotive Industries, Vol.* 77, 1937. Stresses so calculated were plotted against cycles at failure for many heat-treated gears having surface hardness of approximately C60 Rockwell. These gave points that clustered fairly closely about a straight line on a logarithmic chart.

From the average values for gears that had been shotpeened after carburising, a straight line was obtained that lay well above the line for gears that had not been shotpeened. For example the calculated stress for failure at 100,000 cycles was approximately 102,000 p.s.i. for shotpeened teeth against 92,000 p.s.i. for those not shotpeened. At 9,000,000 cycles failure occurred at 74,000 p.s.i. and 52,000 p.s.i. respectively, to show that the advantage of shotpeening is greater when the required life is longer. Conversely, for a calculated stress of 80,000 p.s.i. shotpeening increased the average life from 300,000 to 3,000,000 cycles.

Shotpeening was the last operation on the teeth, and no attempt was made to protect the tooth flanks. The slight roughening of the surface that occurs does not appear to be detrimental, but if desired, a smooth finish can be obtained by using a protuberance hob, followed by hardening, shotpeening and grinding. If this sequence is followed, the tooth flanks can be ground without removing the shotpeened surface in the fillet, where the bending stress is a maximum.

Carburised gears can be allowed a distinctly higher stress than through-hardened and cyanided gears. The process is also effective for gears of lower hardness and for spiral bevels and hypoid gears. It is suggested that a 10 per cent. increase in stress may safely be allowed.

Scoring and Pitting

A method has been developed for calculating scoring resistance in spur and helical gears, on the basis of various assumptions, that gives good correlation with test data. This also assumes that load is uniformly distributed on the average total length of contact lines. The criterion of failure by scoring is the product of PVT, where P is the maximum compressive stress in p.s.i. as determined by the Herz equation for cylindrical surfaces for the point located at the tip of the gear tooth or pinion tooth. It is based upon total tooth-load, average length of lines of contact and the curvature of the tooth surfaces in the plane normal to the line of contact at the selected point. V is the sliding velocity in feet per second of the surfaces at the selected point, and T, in inches, is the distance in the plane of rotation from the pitch point to the selected point.

Test data for fully-hardened gears, lubricated with mineral oil and operating under very varying conditions of

torque and speed, show that failure through scoring generally occurs in gears for which the Value PVT exceeds 1,500,000. Values above this may, however, be used with extreme pressure lubricants. Shotpeening has little direct influence on resistance to scoring, but its strengthening effect in relation to bending permits gears to be designed with finer pitch than would normally be deemed sufficient for given bending stresses.

The finer pitch, although it implies reduced thickness at the root, is advantageous in that, with it, the length of the tooth and the sliding velocity are both reduced. Moreover, with large teeth, and particularly in high ratio gears, the tooth action approaches the base circle of the pinion, and owing to the small radius of curvature of the pinion tooth in that region, high compressive stresses are set up. Insufficient data are available to establish the validity of any method of calculating the compressive stresses that give rise to pitting. A limiting value of 200,000 p.s.i. has been used with success. This value could probably be increased without undue risk of pitting.

Conclusions

These investigations certainly suggest that shotpeening does improve the life of gears. Careful design still remains the chief factor, but shotpeening does allow the weight and size of gears to be reduced. At the same time it also reduces the incidence of failures through both fatigue and scoring. It can also reduce production costs. An increase of only 10 per cent. in the allowable stress for components produced in large quantities may lead to very considerable savings in materials. For example, it has been estimated on this basis that a production machine used for shotpeening large coil springs will save at least 25 dollars per hour of operation.

Machine Tools

IN the course of a discussion some little time ago, an American design engineer, not connected with the automobile industry, stated that he and his fellow designers produced designs that were functionally good, but not necessarily commercially satisfactory. In fact, he went on to say that the commercially satisfactory product was developed only after the original designs had been reviewed and simplified through consultation with the production engineers. This, like many other generalizations, is no doubt an exaggeration, but it does embody a germ of truth.

The exhibition of machine tools recently held in Paris lends truth to the American engineer's remark. Many of the exhibits were undoubtedly the work of highly skilled designers, but they did raise a suspicion that perhaps the designers had been allowed too much latitude. As a result, many of the machine tools incorporated features that are only seldom of real use in production. This, often unnecessary, elaboration arises from two causes. First, the sheer pleasure a designer can derive from solving a difficult and intricate problem. Secondly, and probably the more important, through machine tool users specifying more elaborate machines than are really necessary. It is to be hoped that both tendencies will be held in check in this country.

A shortage of machine tools is already on us, and unfortunately the machine tool industry is not one that lends itself to rapid expansion. Because of this, it is important that users specify the simplest possible machine tools for the applications on which they are to be used. Production engineers have a great responsibility in deciding what types of machine tools will best meet their specific needs.

It is perhaps not sufficiently realized that the use of the simplest possible machine tools, will bring two important advantages. Firstly, it will allow the machine tool industry to raise outputs to the optimum level. Secondly, the simpler the machine the lower the capital cost with a consequent reduction in machine charges and therefore, lower unit costs. In the present state of export markets for almost all British goods, it is almost unnecessary to stress that prices are now a vital matter.

Although it is production engineers who must decide what is the simplest form of machine that will be suitable for a specific application, the design engineers must also play their part. It is not suggested that the production engineer should overrule the designer, but there should be the fullest possible co-operation to produce a design that is functionally good yet one that can be produced as easily as possible. What can be effected through this co-operation is clearly shown in the production of Ford Consul and Zephyr engines, where only one line is required for the cylinder blocks for both engines and only one line for the cylinder heads. The production economies thus made possible are very important.

There is, of course, still a need for both complex and special purpose machines. Even these, however, may be simplified if there is close co-operation between the machine tool manufacturer and the user. Evidence of this is to be found in the machine lines recently laid down by Vauxhall Motors Ltd. and Ford Motor Co. Ltd. These lines include many special purpose machines which are in the main built up from standard units.

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THE "E" TYPE VAUXHALL

A New Version of a Popular Family Saloon

AFTER a run of three years, the Vauxhall Wyvern and Velox range has been redesigned in keeping with what is currently considered modern as regards body style. The familiar scheme of fitting two alternative engines of different sizes to one basic body structure is still retained. It enables both performance and economy markets to be catered for without the overheads that would accompany tooling for two complete body styles. As before, a separate chassis is available for those countries that wish to complete their own bodies.

The four-and six-cylinder Wyvern and Velox engines remain the same, except for a few minor modifications. For example, on the Velox the sump has been modified to clear the front cross member. The engine mounting axis has also been changed, with the result that a new manifold is required so that the carburettor is truly vertical when the engine is installed. On the other hand, the change that has aroused the greatest interest is the adoption of long and short wishbones and coil springs for the front suspension. In 1935, a Dubonnet type was employed, followed in 1937 by a single leading arm arrangement with torsion bars and toggle operated coil springs to produce a variable rate. The semi-floating rear axle has hypoid gearing as distinct from a straddle-mounted spiral bevel pinion that was used previously.

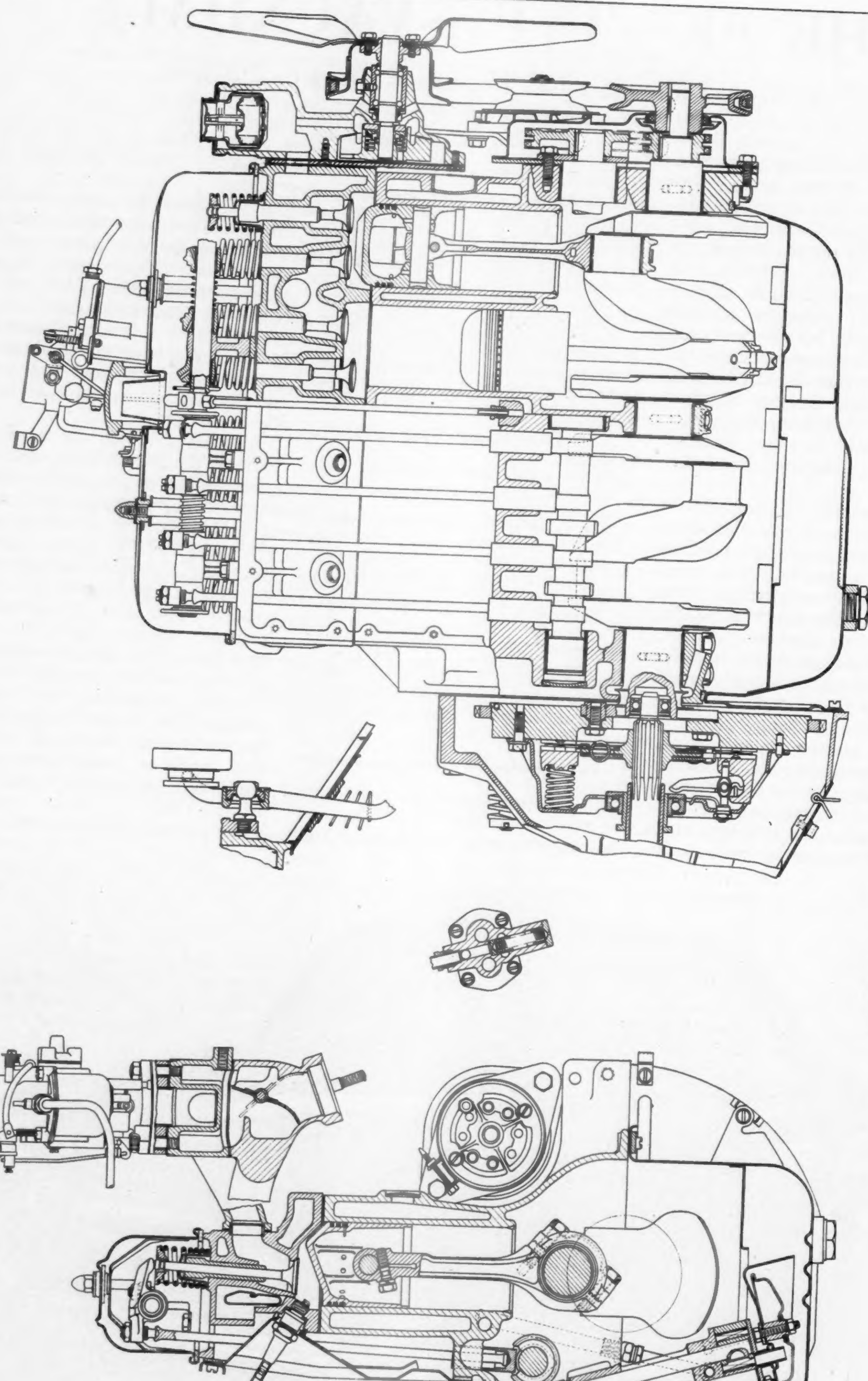
SPECIFICATION

ENGINE: Four cylinders. Bore and stroke 69·5 mm × 95 mm. Swept volume 1,442 c.c. Net b.h.p. 33 at 3,400 r.p.m., maximum b.m.e.p. 116 lb/sq in at 1,800 r.p.m., net brake torque 68 lb-ft at 1,800 r.p.m. Compression ratio 6·4:1. Overhead valves.
CLUTCH: Single disc, 7¼ in diameter. Total friction area 43·3 sq in.
GEARBOX: Three speeds, baulked synchromesh on top and second, steering column control. Ratios, top 1:1, second 1·638:1, first and reverse 3·434:1.
REAR AXLE: Hypoid. Ratio 46·25:1. (8/37).
FRONT SUSPENSION: Independent by long and short wishbones and coil springs.
REAR SUSPENSION: Leaf springs with off-set axle mounting.
BRAKES: Vauxhall manufacture, Lockheed operating cylinders, leading and trailing shoes with one Huck link. Diameter 9⅛ in, width 1½ in front, 1¼ in rear.
DIMENSIONS: Wheelbase 8 ft 7 in. Track, front 4 ft 5 in, rear 4 ft 6¼ in. Overall, width 5 ft 7 in, length 14 ft 4¼ in, height (unladen) 5 ft 3¼ in. Kerb weight, front axle 1,205 lb, rear axle 1,095 lb, total 2,300 lb.

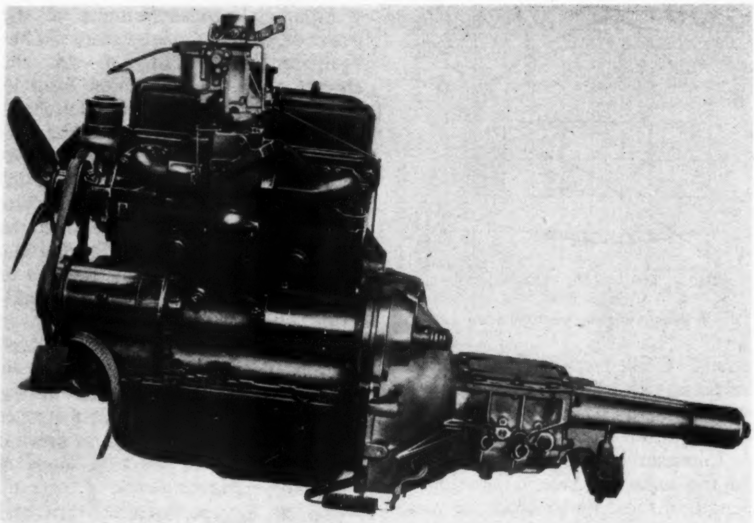
In the following notes, the four-cylinder Wyvern model is referred to in detail, as the Velox power unit was described in the *Automobile Engineer* of November 1949. Most of the remaining components on the "E" type models are similar on both cars but there are, of course, differences in gear and axle ratios and tyre sizes to suit the different engine capacities and wheel loadings.

Rated as a 12 h.p. by the R.A.C. formula, the four-cylinder engine has a bore and a stroke of 69·5 mm and 95 mm respectively, giving a capacity of 1,442 c.c., and a net output of 33 b.h.p. at 3,400 r.p.m. Although the stroke of the four-cylinder engine is 5 mm less than that of the six-cylinder unit, it is by no means a "square" engine, the bore : stroke ratio being 0·73 approximately. The right-hand side of the engine is clear of major auxiliaries, but on it is mounted the by-pass oil filter and also the components driven by the camshaft, such as petrol pump and the windscreen wiper drive mechanism. At the front end of the engine the water pump is attached to the cylinder

On the E type Vauxhall Wyvern the familiar bonnet flutes are retained



GENERAL ARRANGEMENT OF VAUXHALL WYVERN ENGINE
Bore and stroke 69·5 mm. × 95 mm. Swept volume 1,442 c.c.



On the left-hand drive engine the clutch and gear box connections can be clearly seen

block. A single V-belt drives the pump and the fan-cooled dynamo mounted on the left-hand side of the engine. At the opposite end of the crankcase, and almost in line with the dynamo, is the starter motor. Both inlet and exhaust manifolds are on the left-hand side of the cylinder head.

Cast in an iron alloy, the cylinder block is designed with a view to economical production, and the risk of distortion is minimized by maintaining a symmetrical form wherever possible. The water jackets extend almost to the full length of the bores. By splitting the crankcase on the crankshaft centre line, a simple, compact block arrangement is possible in spite of the relatively long stroke. Crankshaft and camshaft are each mounted in three "Thin-wall" bearings, an arrangement which enables one web to provide support for both centre bearings, as distinct from the system used by some manufacturers where four camshaft bearings are employed in conjunction with a three-bearing crankshaft.

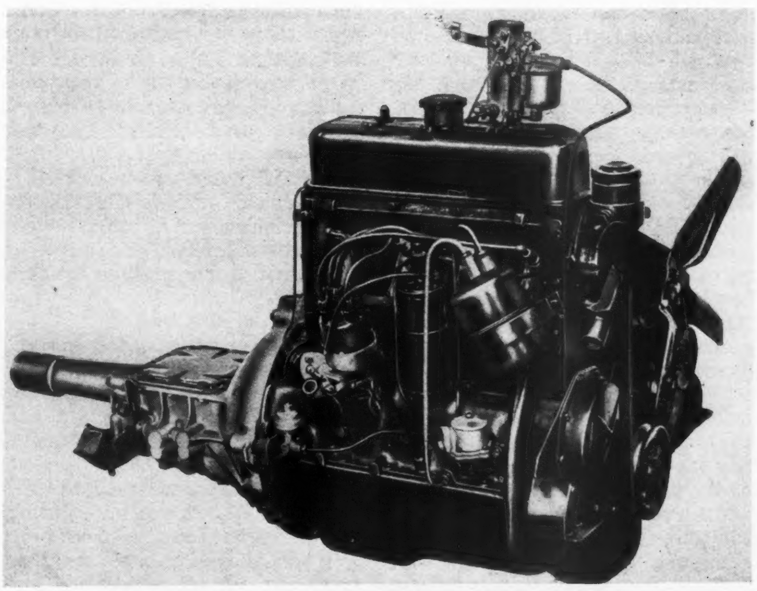
To simplify machining, the main bearing caps are not located in recesses machined in the casting, but are positioned by dowels and secured by set screws. On the other hand, to simplify casting operations, and probably to save weight, despite the expense of a little extra machining, the tappet cover is arranged to seal on both the cylinder block and the head, and is held in place by 16 screws. The sparking plugs necessarily pass through this steel cover and, consequently, cork rings are placed around the sparking plug bosses and sandwiched between the cover plate and the cylinder head to furnish a seal.

Forged from "45" carbon steel, the crankshaft is of sturdy proportions and has integral balance weights. Together with the flywheel and clutch it is dynamically balanced on a special General Motors machine. The front and centre main bearings are 1·8035 in diameter but the rear bearing diameter is increased to 1·9285 in. Bearing lengths, front to rear are 1·3437 in, 1·051 in and 1·7656 in. End thrust is taken by the flanged "Thin-wall" centre main bearing, and no adjustment is provided for end float. The oil thrower ring behind the rear main bearing is integral with the crankshaft and functions in conjunction with a groove machined in the bearing housing. A drain hole leading back into the crankcase is bored in the rear bearing cap.

Flywheel attachment is effected by four bolts and special two-diameter nuts. The smaller outside diameter of the nut extends right through the crankshaft flange to form a dowel location for the flywheel. The larger diameter has a flat machined on it, which contacts a shoulder on the back of the crankshaft flange to prevent rotation during the tightening operation. To provide clearance for the clutch plate springs after the flywheel has been refaced, it is necessary to use rather thin bolt heads, a point that could complicate assembly.

Both the cast iron sprocket on the crankshaft and the steel sprocket on the camshaft are keyed and pressed on to their respective shafts, as is the crankshaft belt pulley. This pulley keeps the front oil-thrower ring in position and its hub provides a contact surface for the felt oil seal which is located in the pressed-steel timing cover.

To facilitate assembly and maintenance, the "35" carbon steel connecting rod big-ends are split at an angle which enables them to be passed through the cylinder bores. The bearing diameter is 1·625 in, with an effective length of 1·247 in. Located by means of coarse serrations, the bearing caps are each held in place by two manganese-molybdenum steel set-screws. As is the case with the main bearings, the big-end bearings are of white metal, steel backed, and are prevented from rotating by the usual notch system. The gudgeon pins, produced from case-hardening, manganese steel, are located and clamped against rotation by pinch-bolts which are fitted into

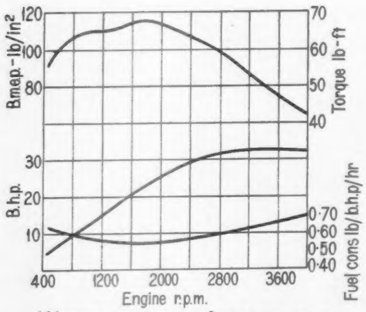


The tappet cover forms a mounting for the coil and the oil filter

the slotted portion of the I-section connecting rods.

The pistons, well ribbed internally, are die cast in RR.53 aluminium alloy. Two compression rings and one slotted scraper ring are employed, the second compression ring being of a stepped or pressure-backed form. The piston head is of an unusual pent-roof form to produce both squish and turbulence in the combustion chamber. To prevent detonation at the ends of the flame travel, a considerable cooling effect is achieved by the ample water passages around the head in the region opposite the sparking plugs. Water cooling is also provided right round the valve seats, and in the case of the exhaust ports, cooling is further promoted by jets from holes drilled in the brass tube that runs the whole length of the cylinder head and through which the coolant supply is circulated by the water pump.

Single springs control the overhead valves. These are quite orthodox, the exhaust valves being produced in chromium-silicon "X.B." steel by the upset forging method. Throat diameters are, inlet 1·0625 in and exhaust 1·0312 in, with valve lifts of 0·2899 in and 0·291 in respectively. With the standard running clearances, the effective valve timing is, inlet opens 4 deg before top dead centre and closes 34 deg after bottom dead centre. Exhaust opens 47 deg before bottom dead centre and closes 4 deg after top dead centre. Pressed steel cups are fitted below the valve springs to retain the felt washers, which rest on top of the cast iron valve guides to control the lubrication of the valve



Wyvern engine performance curves

stems. Chilled cast iron tappets run direct in the cylinder block and transmit motion by means of push rods and rockers, both of "40" carbon steel.

Lubricant is supplied under pressure to the main oil gallery, running the length of the cylinder block, by means of a gear pump driven by an extension of the distributor spindle. A non-adjustable relief valve is held in place by a low-rate spring to maintain an oil pressure of 35–40 lb/sq in. The short tubular intake to the pump is provided with a gauze strainer. Oil is fed to the main journal, crankpin and big-end bearings by suitable drillings in the crankcase and crankshaft. The connecting rod big-ends are also drilled to supply lubricant to the cylinder bores and gudgeon pins.

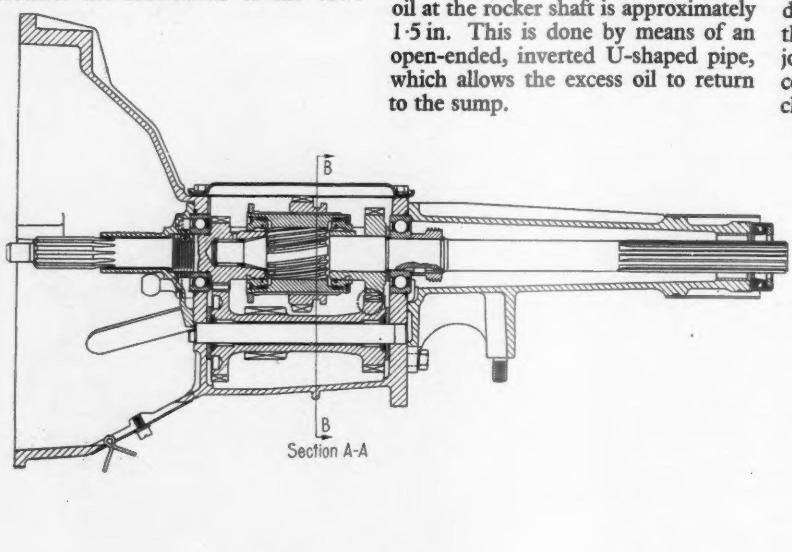
Tappings are provided in the main gallery for the oil pressure indicator, the A.C. by-pass oil filter, and a take-off pipe to the hollow rocker spindle. To prevent excess oil flooding the valve gear, a restrictor and weir system of oil control is used so that regardless of oil pressure, the maximum head of oil at the rocker shaft is approximately 1·5 in. This is done by means of an open-ended, inverted U-shaped pipe, which allows the excess oil to return to the sump.

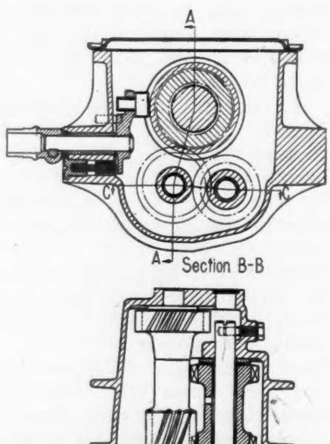
Although basically similar to the previous design, the inlet manifold has been modified slightly to suit the change of engine inclination from 1½ to 5 deg. The hot-spot manifold is retained and the temperature is thermostatically controlled by a bi-metal spring and a balance weight which operate a control flap. The exhaust gases pass right round the bottom of the inlet riser during the warming-up period, but by-pass it completely when the engine is at running temperature. The flap, made of hot-rolled, stainless iron and welded to a spindle of similar material, is mounted in bushes pressed in the exhaust manifold. The outer surface of these bushes is serrated to prevent them creeping or working loose at high temperature.

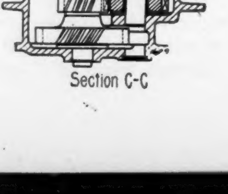
To cut engine assembly time and also to ensure uniform covering, the copper-asbestos cylinder-head gasket is pre-coated by the manufacturers with a plastic lacquer. Distortion of some of the cork gaskets used in the engine has been reduced by the adoption of a paper insert which, in effect, produces a cork-paper-cork sandwich.

Clutch

A standard 7¼ in Borg and Beck six-spring clutch is used and it is actuated by means of a ball-bearing withdrawal mechanism. This bearing is packed with grease and sealed during manufacture and therefore requires no further lubrication in service. The total friction area is 43·3 sq in, and an assembly load of 115-125 lb is used. The clutch withdrawal mechanism is unusual in that the forked lever is pivoted on a ball joint mounted on the gearbox front cover instead of on the conventional clutch cross shaft.







Gears and splines of single helical form are used in the three-speed synchromesh gearbox

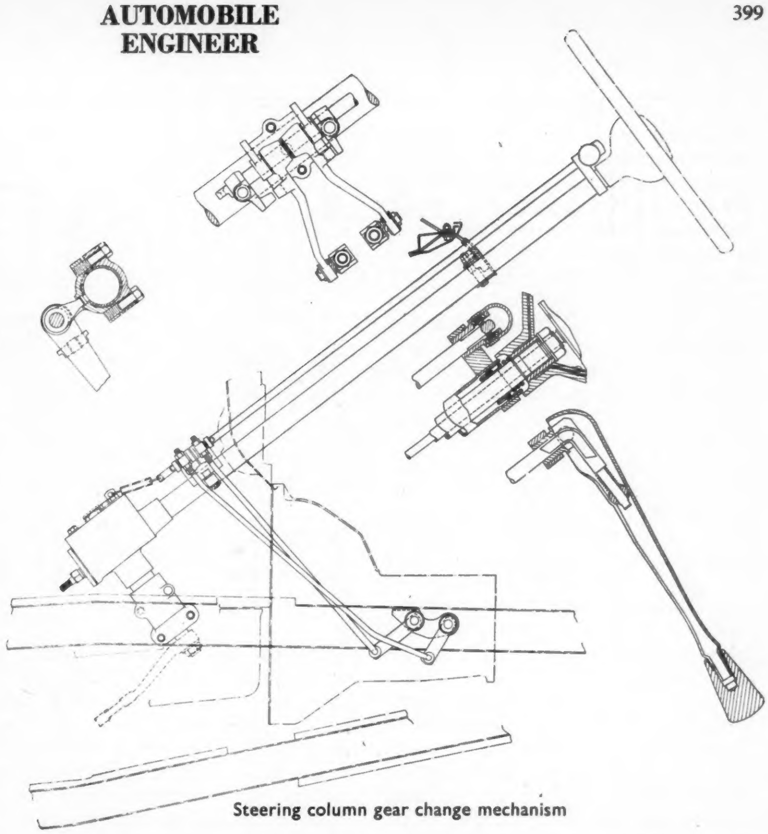
Gearbox

Cast in aluminium, the three speed and reverse gearbox casing and bell housing are produced in one piece. Also made of aluminium, the gearbox rear extension is attached to the main casing by four studs and nuts. The main shaft is supported by a deep-groove ball bearing in the rear wall of the box and, at the end of the gearbox extension, by the sliding end of the propeller shaft universal joint. This component is internally splined to engage with the rear end of the main shaft, and both rotates and slides in a steel-backed, white metal bearing pressed into the end of the gearbox extension. To the rear of this bearing is fitted an oil seal.

The front end of the main shaft runs on needle rollers in the input gear shaft. These rollers are not housed in a cage and, therefore, care must be exercised during assembly. End location of the rollers is provided by a thrust ring at the inner end and a spring ring at the outer end. The input shaft is in turn supported in the front wall of the gearbox by a deep-groove ball bearing, and forward of the clutch by a ball bearing pressed into the end of the crankshaft. Each deep-groove bearing is located by circlips sprung into a groove cut on the periphery of the outer race and seating in a recess in the gearbox wall. The circlips are held in place by the gearbox extension in the case of the rear bearing, and at the front bearing by the cover plate carrying the tubular extension forming the support for the clutch withdrawal bearing. This member also serves to enshroud the oil return thread cut on the input shaft.

All gears are of single helical form and produced from case-hardening, nickel-molybdenum-manganese steel forgings. A baulked synchromesh system is provided on second and top gears. The gear ratios are: top, 1 : 1, second 1·638 : 1, first and reverse 3·434 : 1.

To counteract unbalanced loads that could result from the use of single-helical gears, helical splines are used on the sliding members of the synchromesh unit. Bronze cones are fitted to the inside of the sliding member and engage on steel mating cones on the input shaft gear or the second speed gear. To ensure rapid synchronization and rupture the oil film between the two mating surfaces, the bronze inserts are threaded internally with a fine, V-form tool. Spring detent rings are fitted on the inside of the steel cones and engage on a cam form cut on the splines of the respective gears. The layshaft cluster is mounted on needle rollers which run on a hardened shaft, end thrust being taken by washers placed at each end of the cluster.



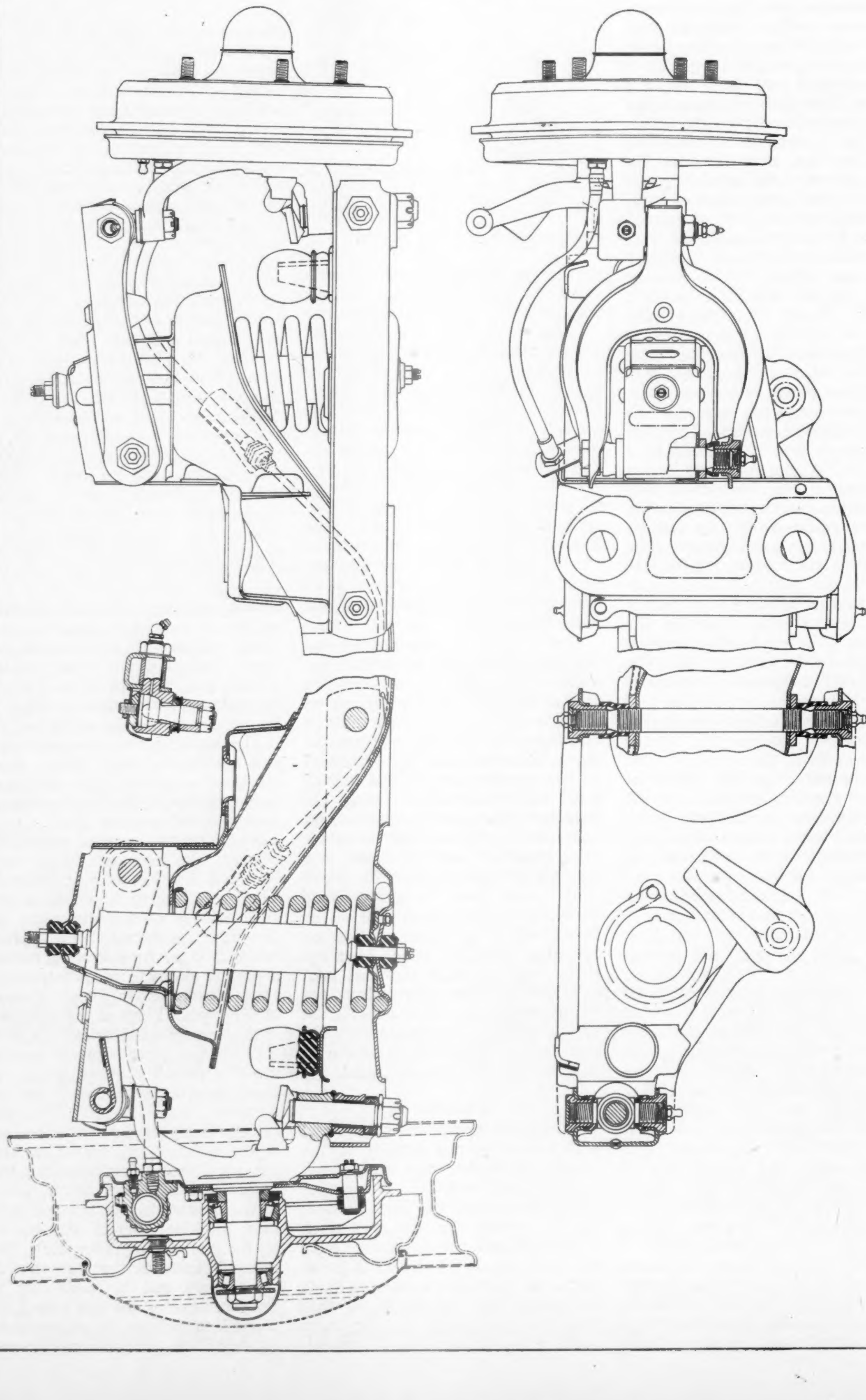
Steering column gear change mechanism

Gear selection is arranged by two spindles mounted side-by-side at right angles to the gearbox centre line. The ends of these rods are suitably formed to engage in the groove on the first speed gear and the ring projection on the synchromesh unit. Discs mounted on the spindles between the striking gear and the gearbox casing are arranged to engage with the selector and interlock mechanism; interlock being provided by a ball sliding in a hole which comes between the edges of the crank discs. To accommodate movement of the power unit on its flexible rubber mountings, the gearbox operating spindles have extension shafts with a form of universal joint at each end. The outer ends of the shafts are supported by bearings in the body structure. The remainder of the gear change mechanism is similar to that used on the previous model.

Rear axle

A semi-floating rear axle is employed with hypoid gearing instead of the straddle mounted pinion and spiral bevel used on the earlier models. A one-piece malleable iron casing is used, and into this are pressed the steel axle tubes. As an additional precaution the casing and tubes are dowelled. When pressing in the tubes care is taken to locate the inner ends accurately, as they form the abutments for the outer races of the taper-roller, differential-housing bearings. The axle tube is, in fact, a composite structure consisting of three pieces, the tube, the spring pad, and the outer bearing housing; all these parts being joined by welding. Projection welding is used in the case of the spring pad.

The hypoid pinion has eight teeth and the blank from which it is machined is produced by the upset forging method out of a case-hardening, nickel-molybdenum steel. It is mounted on two opposed taper-roller bearings, the inner one being 1·5 in bore and 3·125 in outside diameter. The outer bearing is 0·9842 in bore and 2·4409 in outside diameter; the bearing centre distance being approximately 2·75 in. An interesting method is used to assemble these components. First, the cone distance is set by means of shims placed behind the outer race of the inner pinion bearing. A tubular steel distance piece is placed between the two pinion bearings, which are then assembled with the oil seal and propeller shaft flange. The axle casing, with the pinion and bearings is next mounted in a special machine, the whole unit being supported by a fixture which engages on the pinion teeth. Another fixture, also part of the same machine, engages with the nut on the end of the pinion shaft. Both nut and pinion are then rotated at the same speed and by measuring the reaction force on the axle casing, it is possible to measure the torque exerted by the oil seal. After this measurement



FRONT SUSPENSION LAYOUT

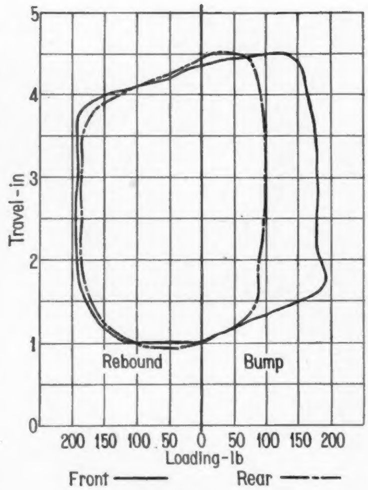
has been taken, the machine rotates both the nut and the pinion, but with a speed differential. This has the effect of gradually tightening the nut, yet enabling pre-load measurements to be taken continuously. An automatic cut-off stops the machine when the desired pre-load is established. Adequate location between the two bearings is ensured by the steel distance piece, previously mentioned, which deforms under a load of 8,000 lb to 12,000 lb.

To enable the pinion nut to be locked at the precisely correct position, the normal method of split pin and castellated nut is replaced by a special nut that has a thin lip on its outer edge. This lip is hammered into a suitable slot in the pinion shaft. The 37-tooth crown wheel is extremely compact and stiff, being only 6·125 in outside diameter. It is registered on the two-piece differential housing, the three components being held together by set-screws. The meshing clearance of the crown wheel is adjusted by means of thick shims placed between the ends of the outer roller races and the abutments provided by the axle tubes. No check on teeth marking is made after the axle is assembled. A measured distance, ascertained previously on a lapping machine, is worked to when the shim thicknesses are ascertained.

The two-pinion differential is of orthodox construction, except that the thrust faces of the differential pinions are flat and no separate thrust washers are provided. Also, the internal splines that accommodate the axle shafts are protected during the carburising process to prevent trouble that might occur if two hardened surfaces were in contact.

Lubrication of the hypoid pinion bearings is arranged by suitable ducts cast in the housing. Oil flung from the tips of the hypoid gear collects between the pinion shaft bearings and then passes through the bearings into the axle casing; directly from the front bearing and by way of a duct from the outer bearing. A deep pressing forms the rear cover of the axle casing and acts as a large oil reservoir. As no breather is fitted, the unit, in effect is completely sealed. To enable the overall dimensions of the differential gears, and consequently the whole axle unit, to be reduced to the absolute minimum, the axle shafts have been kept small in diameter. Accordingly, ground bars of 100-ton steel are used for these items. The shaft diameter on the inner ends is reduced below the root diameter of the splines, which in turn, are of fine pitch to hold down the diameter.

At the outer end of the axle shaft



Shock absorber diagrams

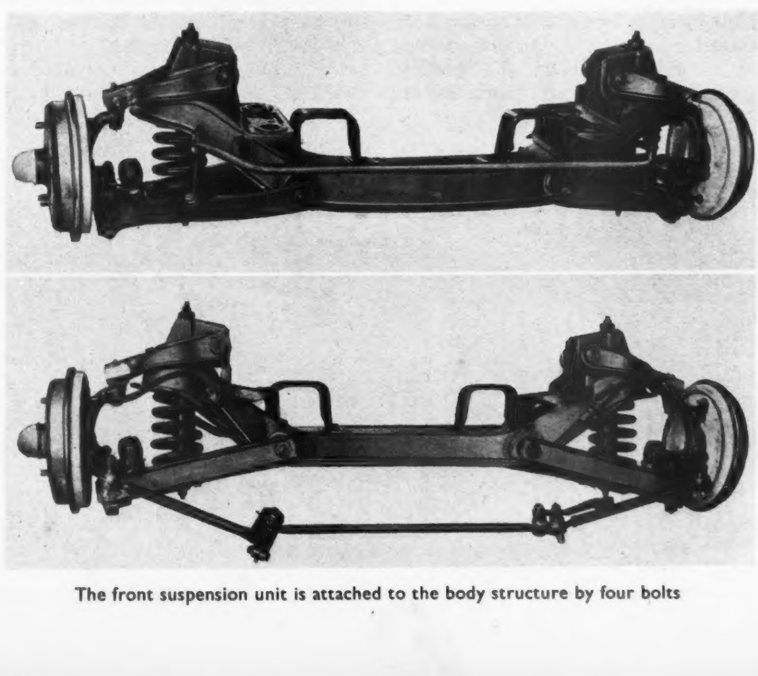
is fitted a combined hub and brake drum. This is keyed and provided with a fitting taper, the nut being locked in a similar way to that used on the hypoid pinion shaft. To prevent a stress concentration, the inside diameter of the tapered bore extends past the tapered portion of the shaft. The axle-shaft ball bearing is of the sealed type and consequently does not require lubrication in service. It is held in place by the brake back plate, which is secured by four bolts. To prevent oil from the rear axle reaching the brake drum, an oil seal fitted into the bearing housing makes contact with a sleeve pressed on the axle shaft. Sandwiched between this sleeve and the bearing is a pressed steel oil thrower ring which aligns with a drain hole, enabling any oil that passes through the seal to escape to atmosphere.

Front suspension

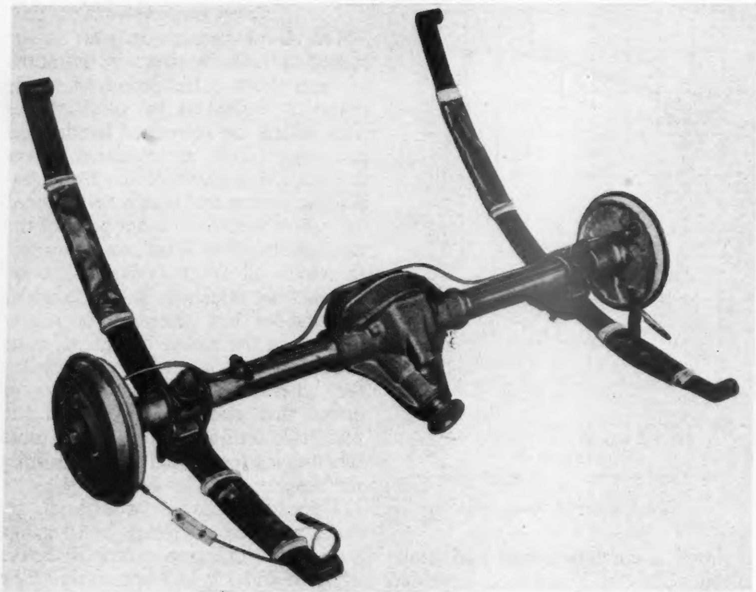
The front suspension unit is attached to the body structure by means of four bolts. Insulation at these points is furnished by stout rubber pads which are restrained inside steel pressings. This arrangement serves to isolate the whole of the front suspension system and eliminates the need for rubber bushes in other parts of the mechanism. The front cross member, to which all front suspension components are attached, is a rectangular pressing of box section, which terminates in the spring abutments at its outer ends. No adjustment is provided for wheel camber, but to prevent errors that could occur due to the material settling, the front cross member is pre-stressed prior to drilling the suspension pivot holes.

Upper and lower wishbones are each one-piece pressings in 10 gauge steel, with effective centre distances of $7\frac{7}{8}$ in and $13\frac{1}{2}$ in respectively. The inner ends of the wishbone pressing are plunged to provide improved support and location for the screwed bushes. In view of the thickness of the material and the dimensions of the plunged portion, it has been found necessary to first stamp out a hole and ream this out to size, prior to the plunging operation, to prevent cracking at the edges. This procedure is adopted on both upper and lower wishbones.

For the top inner wishbone bearings the screwed bushes are fitted into the plunged holes. Although the inner thread forms the actual bearing surface, the outer diameter is also threaded with a special form tool, so that it produces a corresponding thread in the pressed wishbone and remains in



The front suspension unit is attached to the body structure by four bolts



The rear axle is mounted forward on asymmetrical springs

position. The outer ends of the bushes are enclosed to retain lubrication. At the back end, is a plain disc of the Welsh washer type, while the front bearing cover plate is fitted with a grease nipple. The washer is punched with a "size" hole, as distinct from a tapping hole, the screwed nipple is placed in the hole, and the disc placed in a recess in the end of the bush. A form tool then presses on the outside of the washer and distorts it in two directions, outwards to hold it in place and inwards to secure the grease nipple.

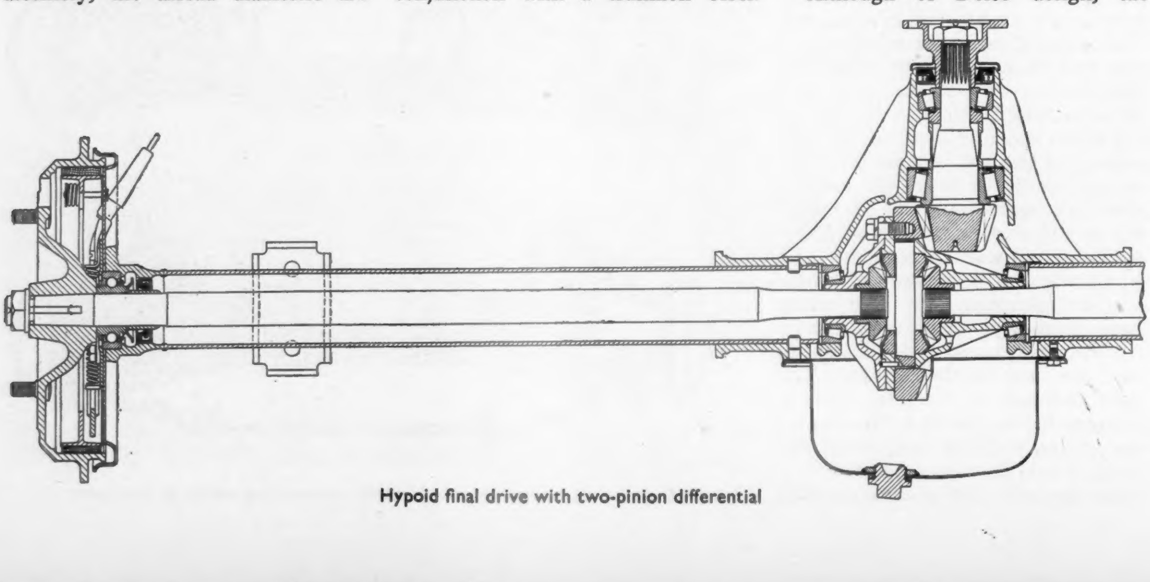
The upper wishbone fulcrum shaft is produced from case-hardening, low-chromium steel and is attached to the cross member by a special obtuse-form thread in a similar way to that adopted on the fulcrum bushes. To facilitate assembly, the thread diameters are larger on one end of the shaft, so that the shaft can be screwed into place when the upper wishbone is in position. The outer top wishbone pivot is a two-diameter ball joint, and is adjustable for wear. A spigot on the joint housing passes through the U-shaped outer end of the wishbone and is held in place by means of a nut. A suitable distance piece is, of course interposed between the channel flanges to prevent distortion, and the end of the spigot is tapped for a lubricator. The lower end of the ball pin is tapered for attachment to the steering knuckle, which is held in place by a nut.

The lower wishbone pressing is attached in a similar way to the upper one at its inner end. At the outer end, the familiar internal-and-external screwed bushes are still employed in conjunction with a trunnion block which enables movement to take place in two planes. A vertical hole accommodates the bronze bush forming the bearing for the shaft, which is pressed into the steering knuckle. The interference fit at this point is important, as the front wheel load is transmitted through this shaft. An interference of 0·0024 in to 0·004 in is called for. As an additional precaution, the top of this shaft is grooved to accommodate a circlip.

At the lower end, below the trunnion block, three thrust washers are fitted. These washers consist of one of oil retaining bronze, sandwiched between two of case-hardened steel. They are held in place by a light pressed steel cover which is peened over to prevent ingress of foreign matter. The whole assembly is locked by means of two nuts, the lower one being split-pinned.

The front shock absorbers and road springs are mounted concentrically, and to prevent rotation the lower end of the spring finishes in a pigtail which locates in a suitable recess pressed in the lower wishbone. This is to ensure that the springs give an even rate on each side, as would not be the case if the springs were free to rotate, as the plane of movement of the lower arm is not in line with the axis of the spring. Again, to ensure even ride, the springs are graded prior to assembly, in three steps of 30 lb/in. The spring has eight working coils and the rate is approximately 250 lb/in, the static deflection being $4\frac{1}{2}$ in. Both bump and rebound buffers are attached to their respective wishbones. In the case of the bump stops, to prevent the rubbers slipping out of their housings when the vehicle is driven over certain types of surface, a grit-laden paint is applied to the underside of the bump rubber contact point on the spring pad pressing.

Although of Delco design, the



Hypoid final drive with two-pinion differential

Suspension attachment points are triangulated to form a stiff structure

shock absorbers are produced by Vauxhall Motors Ltd. and are similar to the American product, with the exception that they are not completely sealed by welding and can be dismantled should overhaul or repair be necessary. However, there is no provision for topping-up. Shock absorber removal can be effected without removing the road spring, but it is necessary to remove the two plates which sandwich the lower spring pad and form the lower shock absorber attachment point. The inner plate is provided with flats that enable it to pass through the circular hole in the wishbone. Shock absorber attachment is quite orthodox, by means of screwed ends and thick rubber washers which are held in place by nuts.

The combined brake drum and wheel hub runs on two opposed roller bearings fitted on the front stub shaft extension of the steering knuckle. Interposed between the inner bearing and the knuckle flange, is a large diameter washer which is deeply chamfered on its inner edge to minimize stress concentration at the point of reduction in diameter and also to clear the radius on the stub shaft at that point. The outer diameter of this washer forms a bearing surface for the felt sealing ring which is fitted into the hub behind the outer race of the inside bearing.

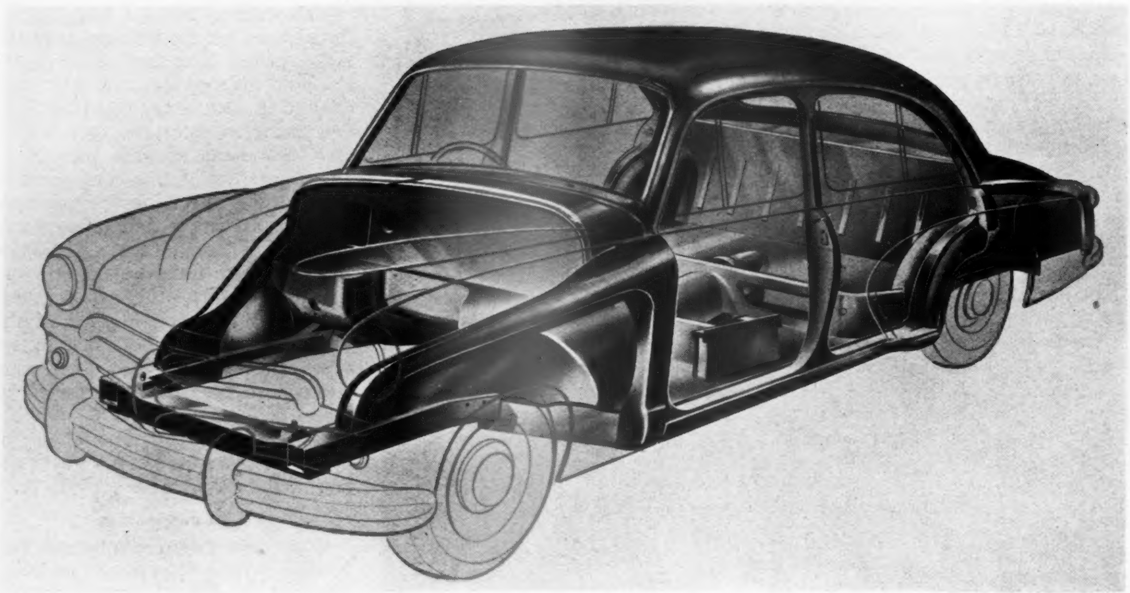
To facilitate machining, the bore for the outer bearing race is machined right through and not stepped to form a shoulder. Instead, it is grooved to locate a circlip. To ensure concentricity, the brake drum friction face is machined after the wheel bearing outer races have been fitted. They provide, in fact, the location for this operation. The hub attachment is by thick washer and nut in the usual way. Once again a castellated nut is dispensed with and locking is effected by a similar method to that used on the hypoid pinion nut. A pressed steel cover is fitted over the outer end of the hub to retain the grease.

The steering knuckle is drilled in three places for the attachment of the brake plate. The two lower holes also hold the steering arms, which are relieved in such a way that they form two contact surfaces, one each side of each bolt hole, so that rocking or uneven seating of this component is eliminated. A normal three-piece track rod is used with a dummy link and bearing similar to the steering box and drop arm fitted on the opposite side of the body structure. An anti-roll bar is mounted forward of the front cross member and is held in two rubber bushings, the outer arms being attached to the lower wishbones by means of short vertical links; rubber pads are used to permit a degree of flexibility. Measured at the wheel, the rate of the anti-roll bar is 38 lb/in.

Rear suspension

Conventional semi-elliptic rear suspension is used, the spring eyes being mounted in flanged rubber bushes. Three spring plates only are used, the distance between the spring eyes is 48 in and the width of the plates 2¼ in. To prevent the nose of the car dipping when the brakes are applied, the axle is mounted 19 in from the front spring eye. As on previous models, greased-packed, fabric gaiters are fitted over the spring plates and are secured by rubber rings and spring caps. Telescopic shock absorbers are inclined inwards towards the centre line of the chassis to increase stability.

The body panels forward of the scuttle are welded to form a sub-assembly



The main body structure

Brakes

Leading and trailing shoe hydraulic brakes are used on both the front and rear of the car, and are operated by Lockheed cylinders. A Huck link system is employed on the leading shoe to improve load distribution and enable the shoe to follow minor discrepancies in the brake drum form. The pressed steel backing plates are reinforced locally at the bottom fulcrum point by means of an additional bracing which considerably stiffens the mounting of the pivot.

Cast in Lepaz pearlitic malleable iron, the brake drums are $9\frac{1}{8}$ in diameter but, to provide a uniform distribution of braking from front to rear, the brake lining width is $1\frac{1}{2}$ in at the front as compared with $1\frac{1}{4}$ in at the rear. The lining thickness is $\frac{1}{4}$ in and the total lining area is 100·65 sq/in. Brake adjustment is effected by two screwed end fittings, which are in turn adjusted by internally threaded rings which bear on the brake pistons. The hand brake operates on the rear wheels only and consists of a single cable and a fairlead bridle which is connected by a lever to the brakecross shaft.

Body structure

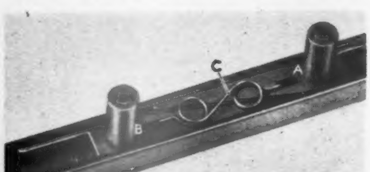
The new body arrangement results in an extremely stiff structure. The box-section members to which the front suspension unit is attached are braced by deep pressings that run up to the sides of the scuttle pressing and so form a rigid triangulation. A similar process is adopted at the rear end, where triangulation is formed by the rear spring anchorage points, the rear seat squab pressing and a bracing member which completes the triangle. As with most integral-construction vehicles, the roof pressing plays an important part in completing the structure, and in this case the fall-away at the rear quarter provides an extremely stiff but light panel and yet enables a large rear window to be used. At the front the spring pillars are quite slender but additional stiffness is no doubt provided by the curved windscreen glass.

In spite of the fact that a hypoid rear axle is used, it is necessary to run a propeller tunnel above floor level in the rear compartment, and this has been formed as part of the structure.

A LOCKING UNIT

JONAS WOODHEAD and Sons, Ltd., Kirkstall Road, Leeds, are introducing a unit by Slide-Lock Ltd., providing for instantaneous locking and release of adjustable sliding fittings. A and B in the illustration are wedges sliding in a channel; a thin, flexible metal ribbon is attached to each end of the channel and runs diagonally across it and between the wedges. Any force tending to close the wedges increases their combined width and locks them against the side of the channel. A force tending to separate them breaks the lock and permits sliding, although the spring C prevents appreciable parting of the wedges.

If the device is used for a half-drop window, a transfer bar is required. This is located between the sliding assembly and the wedges, and has suitably located slots in which engage the pin-like projections on the wedges. Normally, the weight of the assembly is transmitted through the transfer bar and a compression spring to the top of the upper wedge. A sufficient downward thrust, however, compresses the spring until the end of the lower slot in the transfer-plate makes contact with the projection on the lower wedge, breaking the lock and lowering the assembly. A thrust in the opposite direction is transferred to the upper wedge and raises the assembly.



The Slide-Lock unit

The Slide-Lock unit is supplied in two types: the simple Pressure Lock and the Positive Lock incorporating trigger and buttons to actuate the wedges. Advantages claimed are: resistance to vibration and road shock; easy, instantaneous adjustment; automatic locking at any point of travel; smooth, noiseless action; invisible installation; and long, maintenance-free life of constant efficiency. (1982)

SPEED CONTROL

Notes on Heenan-Dynamatic Coupling Applications

IN recent years many previously accepted principles have had to be either abandoned or modified to ensure that full use is made of available labour and materials. This applies both to the production of components and the assembly of finished articles. In the interests of economy and efficient working, new processes have had to be be planned down to the last detail. As was only to be expected, many new problems had to be solved before new production could be started. One of the many needs that arose was that for a variable-speed unit, capable of infinite adjustment and ease of operation, without the complications of a D.C. supply or of expensive and complicated ancillary equipment or switchgear.

With these features in mind, Heenan and Froude Limited, Worcester, have spent many years in developing the Heenan-Dynamatic variable-speed coupling. This not only meets the above needs but also lends itself to many applications in which automatic control and special characteristics are invaluable. These couplings are of the electro-magnetic type in which there is neither mechanical nor fluid connection between the driving and driven elements. The two elements of the coupling are separated from each other by a small air gap and the torque supplied by the primary or driving element is transmitted across the air gap by the action of the magnetic fields linking the two elements together.

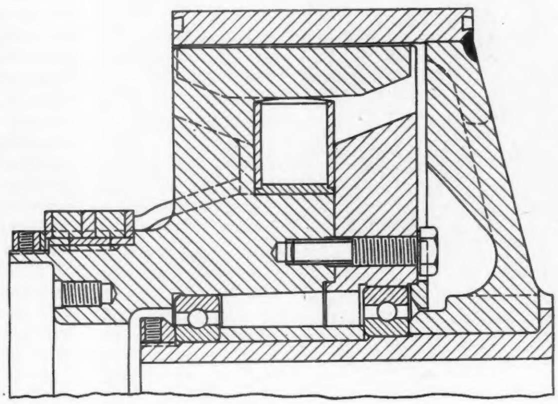


Fig. 1. Half-section of a small Heenan-Dynamatic variable-speed coupling

Coupling construction

A half-sectional arrangement of a typical small size coupling is shown in Fig. 1. This construction is at present used for duties up to 3 h.p. Essentially the coupling comprises two concentric members, the outer being an annular ring and the inner member resembling a pair of side-toothed discs with the teeth extended clawlike so that they interlap towards the centre. The construction is so arranged that an annular space is left between the inner periphery of the teeth and the boss of the rotor to accommodate a field or exciting coil which is wound circumferentially. The leads to this coil are brought out through one side and connected to a pair of slip-rings which are mounted on a boss cast integrally with one half of the inner member, and the current is fed to the coil through simple carbon brushes.

Both inner and outer members of the coupling are made from low-carbon steel that has special magnetic and electrical properties. They may be of forged, cast or fabricated construction. Usually the outer member is arranged as the driver and the inner member the driven. In the case illustrated the coupling is arranged for mounting directly on a standard motor shaft extension, the mounting sleeve supporting the inner or driven member by means of two ball bearings. In some cases it is possible to couple the coupling output member directly to the machine to be driven, but where it is necessary to make use of an Alnico governor generator (as described later) it is usual for the coupling to be supplied complete with a stub shaft and with a pedestal, for supporting this generator.

In common with some other forms of slip coupling this unit has the characteristic that the input and output torques are always equal and, therefore, when transmitting load there is also a certain amount of heat to be dissipated. At any given torque the heat increases as the difference in speed between the input and output members increases. Provision has therefore to be made for dissipating this heat, and for standard units where air cooling is adopted means are provided for circulating air through the gap between the inner and outer members. In the type of coupling illustrated in Fig. 1, blades on the inside of the end bell provide sufficient fan action to cool the unit adequately throughout its normal range. For larger units which have to deal with much greater powers a rather different construction is adopted.

In these larger units the outer member is provided with milled slots which effectively act as a fan to draw air through apertures in the end covers, the air passing over the inner surface of the outer member and being then discharged radially through the slots. All the heat consequent upon slip is generated in the outer member, no heat being actually generated in the inner member carrying the field coil. In service the temperature of the inner member does tend to rise slightly, but this is due to radiated heat from the outer member and to a small extent to the heat generated by the passage of excitation current through the field coil. Excitation current heat can be ignored for all practical purposes, as excitation requirements are very small. The larger types of coupling are normally supplied as self-contained units in pedestals as illustrated.

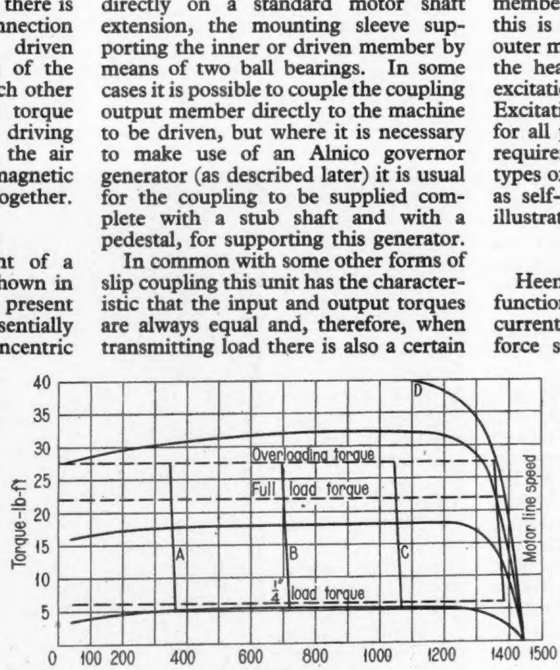


Fig. 2. Speed holding characteristics of a Heenan-Dynamatic coupling
A, B and C are three speed settings chosen at random. They may be at any point in the speed range within the capacity of the coupling. Curve D indicates the torque available under maximum excitation for starting purposes (intermittent only)

Principle of operation

Heenan-Dynamatic slip coupling functioning depends upon the eddy currents created by magnetic lines of force set up by the field coil which makes the inner member an electro-magnet having a number of north and south poles. The strength of this electro-magnet is determined by the amount of current flowing and the number of turns of wire in the coil. When the coil is energized from a source of direct current, magnetic lines of force flow between these north and south poles; by adding a solid ring of soft iron to encircle the poles of these electro-magnets, the magnetic lines of force flow through the iron ring in preference to the surrounding air, owing to the fact that iron

Fig. 3. A drive unit incorporating a Heenan-Dynamatic coupling at Standard Motor Co. Ltd., Banner Lane Works

has a much greater ability than air to conduct magnetic lines of force.

When the field assembly (magnets and coil) is rotated mechanically, the soft iron surrounding it remains stationary until current is applied to the coil. The effect of applying current energizes the coil and in turn causes lines of force to flow through the magnets into the soft iron ring. The mechanical rotation of these magnets produces movement of the lines of force in the soft iron ring, causing eddy currents to flow. These induced eddy currents develop a second field, the strength of which is dependent upon the flow of flux (magnetic lines of force) and the relative speed between the two members. The attraction of these fields makes the ring follow the field in rotation or vice versa as, for practical considerations, the ring is usually the driving member and the field the driven. It will be understood from the above that the essential eddy currents are produced by the relative motion between the two members, regardless of which is driver or follower.

If there is a force tending to retard the motion of the one member so as to prevent it from maintaining the same speed as the other member, the magnitude of eddy currents thus generated increases as the difference in speed becomes greater. This difference in speed is known as slip speed. As the slip speed increases the magnetic lines of force move through the ring materials more quickly, producing more and stronger eddy currents which react with the field magnets to produce more torque at the driven shaft. Similarly, if the current travelling through the coil is increased or decreased (thereby varying the number of lines of force) there will be generated greater or lesser amounts of eddy currents, which in turn affect the torque produced at the output shaft.

It is evident from the foregoing explanation that eddy currents, which in other electrical apparatus are normally considered a detrimental influence upon the efficiency, are purposely created and controlled in the Heenan-Dynamatic coupling, to transmit torques without mechanical contact between the driving and driven members. Thus a major cause of wear is eliminated by this non-friction method of obtaining adjustable speed. In fact, the only wearing parts are the ball bearings, slip-rings and brushes and therefore the coupling can be relied upon to give trouble-free service almost indefinitely with a minimum of maintenance.

The small amount of D.C. necessary to excite the field coil is usually obtained through a small electronic control unit fed directly from a single phase A.C. supply, thus an adjustable-speed drive, providing an infinite variation of speeds is obtained entirely from alternating current. The requirements of a number of different types of drive can be accommodated by varying the coupling characteristics by special circuits in the electronic unit, although the basic requirement of the majority of drives is to provide various output speeds which, when selected, remain constant irrespective of quite wide changes in load.

Conveyor applications

In automobile factories, in common with all other concerns using mass production methods, the conveyors and assembly tracks comprise vital links in the production process. In the majority of cases it is essential to be able to preset a speed, or to adjust the existing speed, and to remain at the selected speed almost indefinitely. With the Heenan-Dynamatic coupling any necessary variation in speed is effected by slipping the coupling the required amount and stepless speeds from maximum down to almost zero under operating conditions are then available.

In the case of a simple conveyor with only one drive unit it is usual for the coupling to be mounted between the driving motor and the main reduction gearbox. The layout of the whole driving unit is very simple—motor, coupling and gearbox—and therefore the efficiency at maximum speed is greater than when using previous methods of speed variation involving a number of gearboxes and belt or chain drives. Maintenance costs are also appreciably reduced. As there is no mechanical connection between the input and output members an extremely smooth take up of the drive is obtained with a corresponding reduction in stresses and elimination of shock loads.

The electronic unit providing the necessary D.C. current to the coupling coil is of sound construction and has proved in practice to be remarkably reliable and free from trouble. It is normally mounted in a ventilated sheet steel case suitable for wall mounting, and utilizes mercury vapour grid-controlled rectifier valves. The D.C. output from these valves is modulated by the output voltage of a small Alnico permanent-magnet governor generator fitted to the coupling output shaft. The output of this governor, the voltage of which varies directly with speed, is fed via a transformer and separate rectifier valve to the grids of the main rectifier valves. This circuit is so arranged that should the output speed of the coupling tend to rise by even a very small amount due to a decrease in load, the increased output of the governor (which is suitably amplified in the transformer—rectifying valve stage) is sufficient to over-bias the main rectifier valves and to cut off the flow of current through them. Excitation thus being removed from the coupling, the output speed will fall minutely until the original preset speed

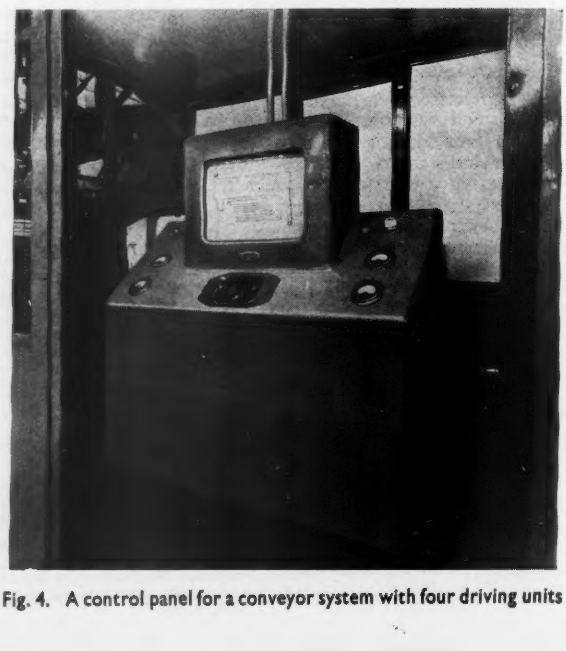


Fig. 4. A control panel for a conveyor system with four driving units

is again reached, when the valves will again pass current.

Conversely, if the output speed of the coupling tends to fall due to an increase in load, the output of the governor falls in sympathy, allowing the main rectifying valves to pass more excitation current to the coupling coil until the original speed is substantially restored. As the electronic governing action is extremely rapid in practice it is impossible, by normal methods of observation, to detect any appreciable change in the output speeds of the coupling on conveyor drives, despite comparatively large load variations. The exact degree of governing can be adjusted in the design to suit requirements. Where it is necessary for extra close speed holding, this can be obtained by arranging a specially designed circuit. The normal standard governing is to within $\pm 0{\cdot}5$ per cent at 1,500 r.p.m. Fig. 2 illustrates graphically this speed holding characteristic under loads which may vary from approximately one-quarter normal load torque to a resonable overload. This method of control is known as "constant speeding" and although it has been described in relation to conveyor drives it is of course suitable for many other applications requiring similar characteristics.

It will be appreciated that as the only connections between the excitation unit, coupling and speed controller are electrical, the speed control knob can be arranged for remote mounting in any convenient position, the speed being selected solely by the operation of this knob. The speed may be adjusted whilst the conveyor is working and if necessary a speed indicator calibrated in units per hour, feet per minute, etc., can be provided so that adjustment of the speed can be accomplished simply, quickly, and accurately.

A Heenan-Dynamatic coupling lends itself to the incorporation of various safety devices. For example, a simple torque-control circuit ensures that the driven machinery is accelerated to full speed in the minimum time consistent with avoiding any excess of motor torque beyond a pre-set but adjustable value. By an extension of this principle, it is possible to arrange that when the load on the conveyor motor reaches any predetermined value the excitation is removed from the coupling itself and the motor is also shut down. This prevents the conveyor from being overstressed and gives closer protection than is possible with the usual shear pins. However, in many applications, the initial starting torque of the conveyor system is in excess of the safe working torque, and to enable the conveyor to accelerate up to speed, the torque-limitation feature is provided with an automatic delay which keeps it out of circuit until the conveyor is up to speed and working under normal conditions. The above refers to simple single-drive conveyors but when used with conveyors having two, three, four or more drive units due to their excess length, the coupling provides many features which have not hitherto been obtainable with conventional forms of drive.

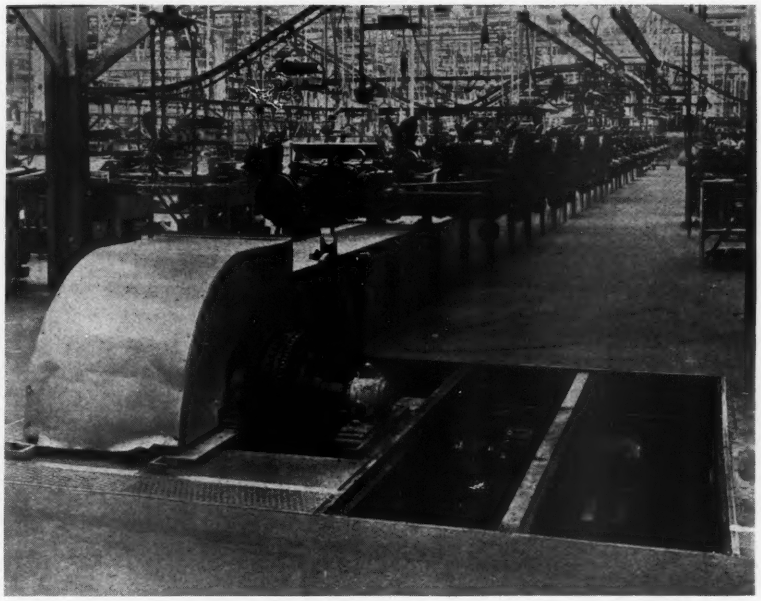


Fig. 5. The drive unit for the assembly conveyor at Standard Motor Co. Ltd., Banner Lane works

To feed components or unit assemblies to the main production line the overhead conveyors employed must frequently be of considerable length in order to serve the extensive floor area. Similarly on process work, such as painting, the conveyor must, of necessity, be routed to pass through spray booths, drying ovens, inspection bays, etc., thus including several strands, bends, changes of level, and varying temperature conditions. Such conveyors are invariably fitted with two or more similar drive units for reducing to an acceptable level the drawbar pull on the chain links. It will be apparent that the life of the chain will be extended if a steady value of pull is applied at all points of the conveyor, which can only be achieved if the several drive units all share equally in the work. The balancing of the total load between two or more drive units, frequently situated a considerable distance apart, has however been a major problem, particularly when the speeds of main and secondary conveyors need to be varied simultaneously and frequently (for instance as between day and night shifts) to suit production needs.

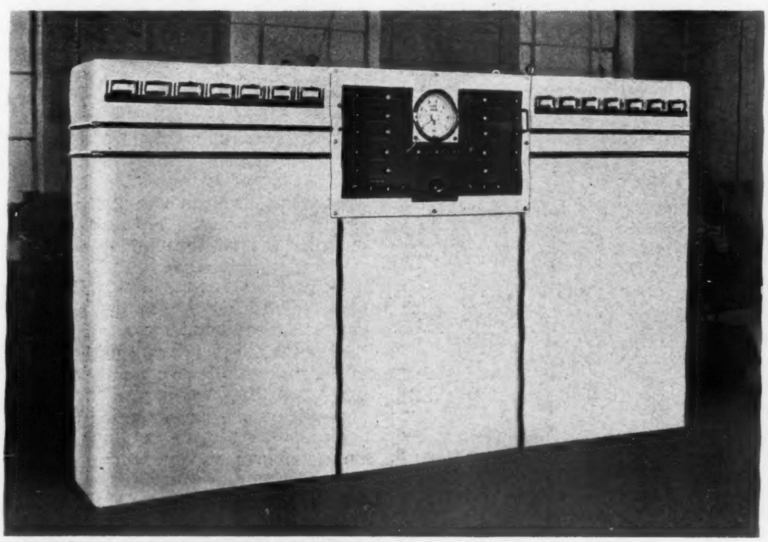


Fig. 6. A special combined control panel for a conveyor system that incorporates several Heenan-Dynamatic couplings

Multi-drive conveyors

The introduction of the Heenan-Dynamatic coupling to this type of

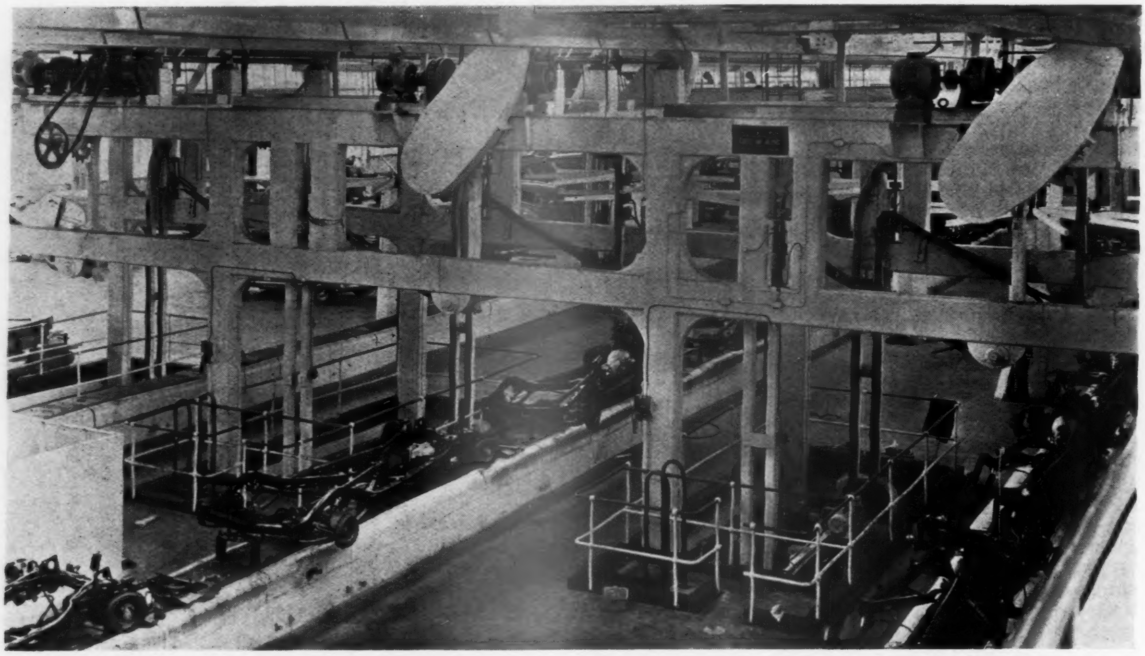


Fig. 7. Drive units installed in the Austin assembly factory for synchronizing movements of elevator units and assembly tracks

installation has provided a straightforward solution to the complex problem. As many as four drive-units on a single conveyor have been balanced for load, and the chain given even tension at all points, while infinite variation in speed on all units simultaneously is centralized at one control station. Each drive unit requires only the driving motor, the coupling and the fixed reduction gear driving the conveyor chain either by sprocket or caterpillar as mentioned above. On a given conveyor the couplings are identical save in one feature; that is to say, one is chosen to be the "master", operating with an inbuilt governor generator, while the remainder are "followers" not provided with governors. The "master" coupling fitted with a governor controls the speed of the system as a whole, each coupling being fed from one common excitation unit, so that each of the "followers" automatically keeps in step with the "master". This equalizes the driving load, reduces the stresses, and eliminates "hunting".

One important result of the effective load sharing is the change in the function of the conveyor chain tension units. These, usually of the weight-loaded type, are no longer required to compensate for uneven pull in the various strands of the conveyor chain. On the contrary they are employed merely to take up increase in conveyor chain length due to wear. Except when being adjusted for this purpose, the tension unit remains static and there would appear to be advantages in the substitution of the weight-loaded type for the slide-rail pattern which may be locked after satisfactory setting has been made.

Fig. 3 shows one of the drive units on a multi-drive conveyor in use at the Banner Lane Works of The Standard Motor Company (Ferguson Tractors). This conveyor is loaded with various components which pass through a bath of primer paint and are next taken through the drying ovens and then on to the unloading bay. This system has four driving units and Fig. 4 shows the control panel, on which is reproduced a diagram of the run of the whole conveyor and the position at which the drives are connected. The panel carries four motor ammeters which enable a check to be taken that each drive unit is doing its share of work. It also carries the single main speed-control knob for adjusting the speed of the system. The drive, including the coupling, for an assembly conveyor in the Banner Lane Factory of Standard Motor Co. Ltd. is shown in Fig. 5.

In the case of complete factory systems where a number of conveyors have to feed one or more assembly tracks, it is frequently essential that each separate conveyor or track should work in synchronism with all the others, within close limits. This is necessary in order that at the end of a normal working period no one of the conveyors is behind or ahead of any other. In most cases where these complete layouts are adopted means of varying the speeds are necessary, firstly for the initial trials, and subsequently depending upon the number of operatives available on any particular shift. With a Heenan-Dynamatic coupling driving each conveyor and track, after the required speeds are matched with individual trimmer potentiometer controls, it is only necessary to operate the single "master" control knob to increase or decrease the speeds of the whole system, simultaneously and in syn-

Fig. 8. Rear axle testing unit, Morris Motors Ltd. (Tractor and Transmission Branch) Birmingham

chronism, from the central control desk.

A number of such complete systems have been, and are being, installed, and Fig. 6 shows the arrangement of a panel for a typical system of the kind described, which is being erected at the works of an important motor car production firm in Coventry. The indicator is calibrated in cars per hour, and it is therefore possible to forecast the production of cars over any set period and to modify this as may be necessary. The efficiency and dependability of such installations has been proved without doubt over a number of years, and losses in production time due to resetting speeds, recalibrating units, etc., have been considerably reduced compared with previous systems.

Synchronized elevation

In the new assembly buildings recently opened for the Austin Motor Company many complex technical problems had to be overcome before the plant as a whole could operate satisfactorily and some of these points were dealt with in the *Automobile Engineer* for September, 1951. Mention was made of the part played by the Heenan-Dynamatic coupling in the various component elevators and Fig. 7 shows these in position on the main framework. Owing to the impossibility of calculating in advance the effects of the various factors involved, it was necessary for many adjustments to be made, and after experiments with conventional means of control had given disappointing results, these couplings were specified. They offered the flexibility and precision necessary to enable the elevators to synchronize so closely with the assembly track that the components would be lowered into position on the moving chassis within limits of a fraction of an inch, timed to within a second or so.

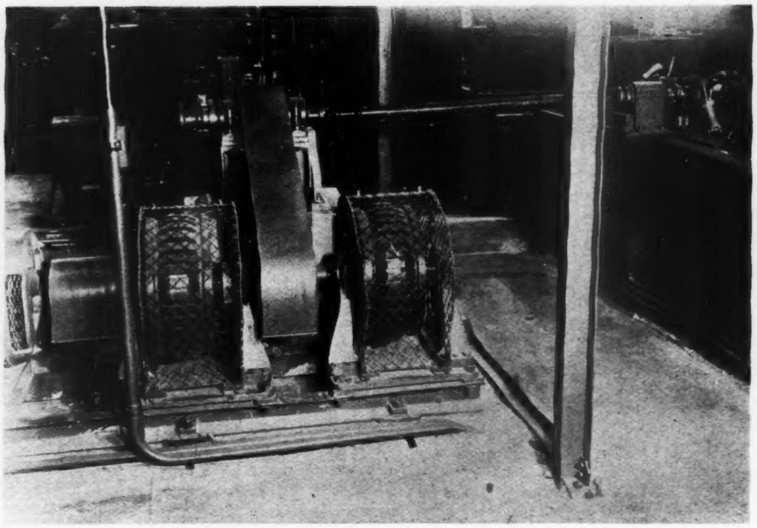


Fig. 9. Couplings incorporated in the drive for the test unit shown in Fig. 8

The elevator operating cycle is initiated by each oncoming chassis on the assembly track making an electrical contact as it reaches a certain position. From this electrical signal onwards the time taken to raise each component from its underground station and place it on the chassis must be strictly controlled. The problem resolved itself into satisfying the following requirements :—

(a) On receiving a signal from the oncoming chassis requiring for example an engine, the engine elevator was to be set in motion.

(b) The engine on its carriage must be lifted from the underground siding with maximum acceleration to a point near the top of its travel.

(c) Its speed must then be reduced to a value which will not set up swinging when passing round the bend to change the direction of travel from the vertical to the horizontal.

(d) Engine and carriage must then travel round a further bend and down to a point directly above the chassis line, and be braked as sharply as possible. Here they must remain poised until the moving chassis sends out another electrical signal.

(e) On receiving this signal the carriage must be lowered until the engine gently meets the chassis. This must be very accurately timed as the clearance between engine and

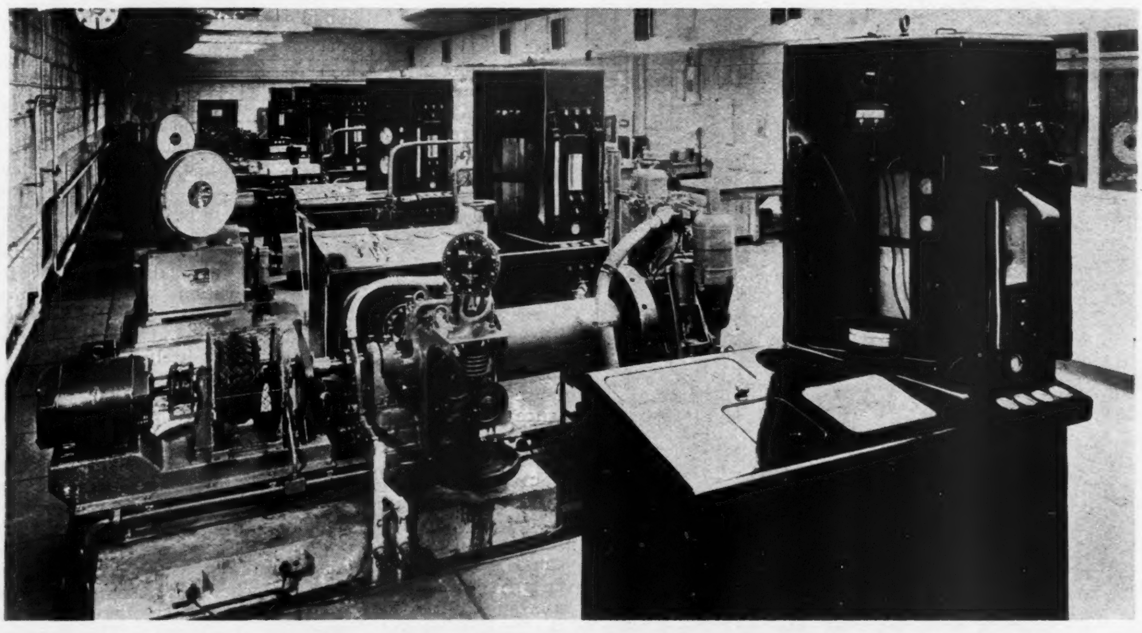


Fig. 10. C. C. Wakefield Ltd. engine testing laboratory. Heenan and Froude couplings are used in conjunction with fixed-speed A.C. motors

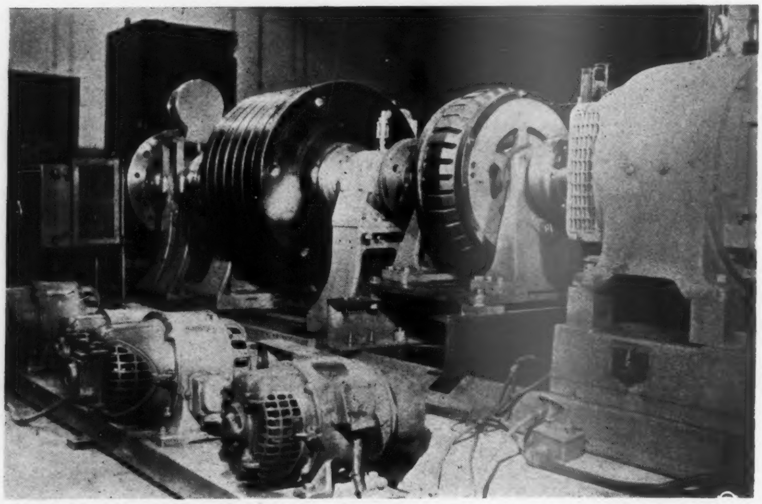


Fig. 11. A Heenan-Dynamatic coupling incorporated in a machine for testing Ferodo brake linings

certain parts of the chassis while falling is only ⅛″.

(f) After the engine hooks have automatically disengaged themselves the above steps must be repeated (but at higher speeds) in reverse, so that at the end of the cycle the elevator carriage is again below ground carrying the empty engine sling which is removed automatically by the next available engine as it is indexed into position on the elevator in readiness for the next cycle.

Obviously, conventional methods were hardly likely to have the flexibility required to permit the various accelerations, speeds, decelerations, etc., to be adjusted one by one until the whole cycle was correct, but the electronic control of the Heenan-Dynamatic coupling was found to meet the requirements, and to maintain them indefinitely when set. The couplings themselves were standard units, some being used in their ordinary capacity as torque transmitters and others as variable-torque brakes or retarders. The time cycle originally set up was one complete operation on the elevator every 2·4 minutes, but this is naturally variable as required, by adjusting the rate of feed of the chassis on the assembly conveyor.

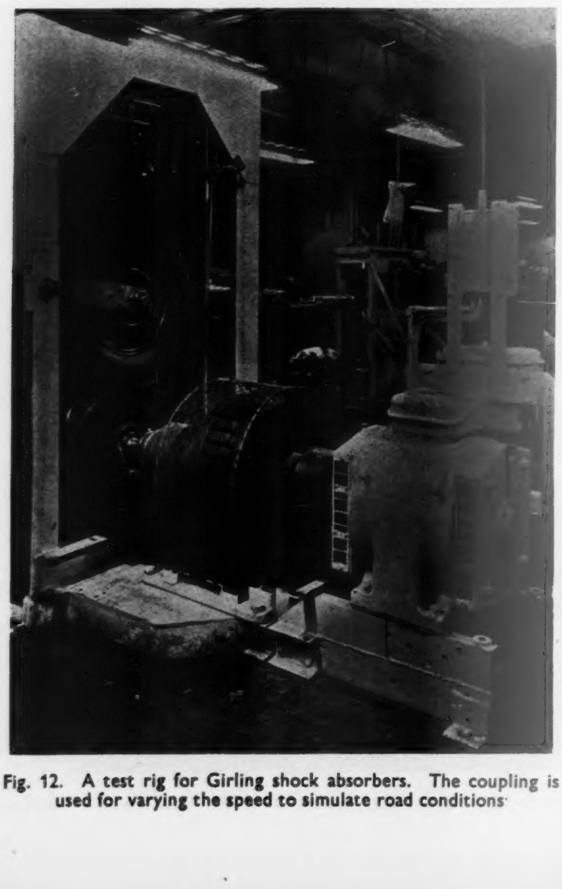


Fig. 12. A test rig for Girling shock absorbers. The coupling is used for varying the speed to simulate road conditions

Other applications

Conveyors and similar drives have been dealt with at some length because of the importance of these units in a mass production system, but the number of applications on which these couplings have been and can be used is almost infinite. Many machines allied to the production and testing of automobile components and assemblies make use of these drives, some of the most interesting ones being briefly described below.

In the manufacture of cylinder liners which are produced on a centrifugal casting machine, it is necessary to be able to run the machine at one speed during the pouring operation, and then to increase the speed to a preset figure for the actual centrifuging of the metal during the casting operation. It must be possible to vary the casting speed depending on the diameter of the mould, and, for speed in production, the cycle should be as rapid as possible. These requirements have been achieved by using a coupling in conjunction with a normal standard A.C. motor, the cycle being initiated by means of push-buttons whilst the motor remains running the whole of the time. In the manufacture of batteries, the coupling has been used with success in driving plastic extruders for the preparation of battery cases, and also in driving rumbler barrels for processing some of the raw materials used in the preparation of battery constituents. Various other types of rumbling barrel for cleaning small components, etc., have been equipped with coupling drive with success.

In the field of mechanical handling, cranes incorporating couplings in the drive give performances as good as, or better then, those obtained with D.C. control, with the added advantage that in this case normal constant speed A.C. motors are used and the control gear is consequently very much simplified. As the motor is running continuously in one direction only, it is running in better conditions than in previous practice where it was necessary to start, stop and reverse the motor frequently. With couplings in the drives of the various motions, "snatch" is eliminated and it is possible to position the load more accurately and smoothly.

To obviate the possibility of defective components fitted in a car being only discovered during road trials, it is becoming the practice to bench-test the various transmission units before they are passed to the assembly lines; in most cases such test rigs need special characteristics to cater for the procedure involved. One of the more interesting test rigs in use is a plant for the running of complete rear-axle assemblies for noise investigation, so that any that exceed the required standard of noise can be rejected and rectified before fitting to the car itself.

A test rig of this type which was built for Messrs. Morris Motors Limited (Tractor and Transmissions Branch) Birmingham, is illustrated in Figs. 8 and 9. Several of these are now in continuous use in the factory. The function of this plant is to accelerate the axle up to the required speed under similar conditions to those applying when the car is accelerating on the road, and when this maximum speed is attained to decelerate the axle under overdrive conditions. The drive to the axle itself is by means of a constant-

speed A.C. motor through a coupling so that the unit may be brought up to speed at the desired rate against the inertia of flywheels to create the desired load. During the second part of the cycle the excitation from the driving coupling is transferred to a second coupling arranged to act as an eddy current brake. As a result, the load is reversed, the drive for the deceleration period being provided by the energy stored in the flywheels, which thus drive back through the axle. The whole cycle takes approximately two minutes, and providing the axle is quiet throughout the test it is passed for final assembly. Heenan-Dynamatic couplings are used in this test plant because of their convenience and simplicity, and for the automatic features which can readily be provided when electronic control gear is used. A similar system has been applied to gearbox test plants where drive and over-drive conditions are required, together with the need for variable speeds, to test the box throughout its complete speed range in each gear.

In the case of engine running-in and testing plants, the use of a coupling in conjunction with a normal fixed-speed A.C. motor has allowed exceptionally smooth and gradual acceleration of the engine, with speed variation up to the required values, both for running-in and starting purposes. Fig. 10 shows a typical test shop arrangement utilizing these units at Messrs. C. C. Wakefield Limited.

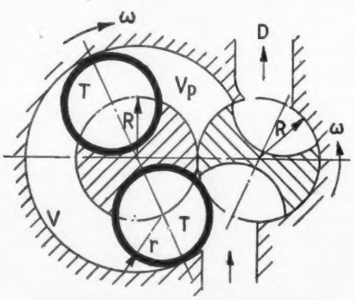
Component test plants have also been fitted with coupling drives, one of which, in the works of Messrs. Ferodo Limited, for testing brake linings is shown in Fig. 11. In this machine it is necessary for a number of flywheels to be brought up to a predetermined speed in a certain length of time, the drive then being removed and the brakes applied, bringing the flywheels to rest again. This cycle has to be constantly repeated for long periods. The use of a coupling in the drive allows the motor to remain running continuously, thus requiring the simplest of control gear. In addition the coupling itself acts as a very smooth clutch free from mechanical wear. The running speed is modified as required to suit the particular test, and acceleration rates are also varied. These features are obtained successfully. The control of acceleration rate is catered for by means of a variable torque-limiting circuit which regulates the amount of torque that the motor is allowed to produce, thus setting the time of acceleration to suit the particular flywheel inertia in use.

A rig for testing shock absorbers to destruction is illustrated in Fig. 12. In this application the coupling is used for varying the speed to simulate road conditions. This plant is used by Messrs. Girling Limited, Birmingham. These notes have dealt only with the use of Heenan-Dynamatic variable-speed couplings in the automobile and associated industries. It will, however, be appreciated that their possible applications are far wider and cover practically every field where variable speeds and automatic features are required.

A MODIFIED ROOTS BLOWER

AN article entitled "A Modified Roots Blower" by A. Baumann in *Schweizerische Technische Zeitschrift*, No. 5, 1951, (*Engineers' Digest*, June, abstract), suggests that the normal design of Roots blower has several disadvantages. The machining processes required to manufacture the two impellers are complicated and expensive, and the efficiency is low since the pumped volume is compressed, not inside the blower itself, but merely by connection with the discharge channel. Finally, the ensuing sudden rise in pressure creates noise. The suggested modified design, see illustration, avoids these disadvantages.

Two solid cylinders of radius R and centre distance 2R, geared to rotate together at equal speeds about their respective axes, each have two large semi-circular recesses of radius $r < R$ cut in the longitudinal direction on opposite sides of their axes. The centres of the recess circles lie on,



Modified design of Roots blower

or near, the original circumference of the solid cylinders. Two circular tubes of outside radius r are fitted closely into the two recesses of one cylinder, whilst the recesses of the other provide clearance through which the tubes may pass on rotation. This "clearance" cylinder, though still pumping a small fluid volume without pre-compression, acts in the manner of a rotating valve. The discharge channel is so placed that the fluid carried in the interspace between tubes and external wall, is pre-compressed before the edge of the rotating valve opens to the discharge channel; the discharge volume V_p being smaller than the suction volume V. The tubes T are slightly undercut to avoid interference.

Only simple machining operations are needed for the manufacture of all the parts. Interference may occur between the tubes and the edges of the rotating valve, but is easily avoided if the interfering metal of the tubes is removed by a flat longitudinal milling cut. The pre-compression of the fluid inside the blower eliminates most of the running noise, increases efficiency, and permits application of the modified design to higher discharge pressures than are usually expected from the normal Roots blower. (1979)

Light Alloy Coach Bodies

ALUMINIUM coaches were among the products of Thomas Harrington, Ltd. which were demonstrated recently to the Press at the Sackville Works, Hove, Sussex. The demonstration was organized jointly by the coach-builders and the British Aluminium Co. Ltd., who supply the aluminium alloy sheet and extrusions used in the construction of the B.O.A.C. and Sarciada Y Pacchiotti (Uruguay) coaches made by Thomas Harrington, Limited.

By using aluminium alloy extrusions and pressed sections, thereby eliminating nearly all of the timber previously used in similar coaches of composite construction, a reduction in weight of about 15 cwt has been made. The manufacturers state that their customers have obtained, as a result, a 10 per cent reduction in the rate of fuel consumption. Economies of this order are, of course, of considerable interest to operators in countries where fuel is expensive. Furthermore, the elimination of timber from the structure is a very necessary feature of vehicles intended for the overseas markets.

Bolts with self-locking nuts, "pop" rivets, mushroom and snap-head rivets are used in the construction, and a certain amount of panel-beating and gas welding are resorted to. Zinc chromate paste is used at joints where dissimilar metals are in contact. The importance of this precaution is not as widely appreciated as it might be. Actually moisture in joints between aluminium alloys and other metals can lead to very rapid deterioration due to the electrolytic corrosion. (1984)

SURFACE HARDENING

High Frequency Induction Heating Equipment for Treating Shafts

Birlec high frequency induction heating equipment for surface hardening shafts.

AN interesting development in high frequency induction heating equipment is shown in the accompanying illustration. It is a vertical shaft hardening machine manufactured by Birlec Ltd., Tyburn Road, Erdington, Birmingham. The high frequency power is obtained from a valve oscillator unit rated for a continuous output of 25 kW at a frequency of approximately 350,000 cycles per second. To keep the high frequency connections to the shortest possible length, the valve oscillator, with its associated high-tension transformer and rectifier equipment is housed in a sheet metal cubicle that stands directly at one side of the hardening machine.

The machine is built in a robust fabricated steel framework, designed to give great rigidity and built with the front sloping backwards towards the top at a slight angle to the vertical. A rigid, large diameter guide post, on which a carriage can run up and down, is mounted parallel with the front. To prevent rotation of the carriage about the main guide post axis, the carriage has an outrigger arm fitted with bearing pads that embrace a second parallel guide member. A forward projection from the carriage supports a stem, on the head of which rests the lower end of the shaft to be hardened. The shaft itself lies in a vee guide on the front of the casing, and as the carriage descends on its guide member, the shaft follows under its own weight.

Drive to the carriage is taken from a variable-speed D.C. motor through a reduction gearbox and a duplex roller chain. The speed of the motor is controlled automatically by an electronic regulator. As a result, the rate of downward travel of the carriage is kept constant within very close limits irrespective of variations in main supply frequency or voltage. The rate of travel can be adjusted by means of a calibrated dial to suit work of any diameter between $\frac{3}{8}$ in and $1\frac{1}{4}$ in. A relatively high speed is used to return the carriage quickly to the top position.

Interposed between the carriage and the vee guide is the high frequency inductor. This is of special design to facilitate rapid change-over from one size to another. The inductor is held rigidly in position by a quick-action cam, and a wide range of different inductors, each designed for a specific component, can be interchanged rapidly and easily. A plug gauge is used to ensure correct alignment with the vee guide. The inductors are water-cooled, and quickly detachable water connections are provided. The electrical connections are made automatically by clamping the inductor in position.

A limit switch actuated by a cam plate on the side of the carriage is used for setting the lower limit of carriage travel. The cam is located by dowels and is held in position by knurled nuts for easy interchangeability. It is also profiled to operate a second limit switch that controls the application of high frequency power to the inductor when only certain parts of a shaft are to be hardened. Thus a cam plate may be cut to suit a specific application, and this in conjunction with the appropriate inductor and motor speed setting, completely defines the heat treatment conditions. This ensures consistent repetition results. Changeover from one type of shaft to another can be effected in about ten minutes. Because of this, the equipment can be used efficiently for relatively small batches of different components.

As the heated work emerges from the inductor it is automatically quenched by water spray. Water safety relays are provided to prevent the application of power until the inductor cooling water and the quenching water are turned on. Hinged access doors are provided, one at the side for changing cam plates and the other at the rear for general servicing. They are electrically interlocked to make the interior dead when either door is opened. All the operating controls are at the front of the machine. In normal use only two controls are required, the main switch and a foot switch that is used to initiate each cycle. Once the work is placed in position and the foot switch is depressed, the cycle is fully automatic. That is, the work is taken through the full hardening cycle and then returned to the unloading position. All setting-up adjustments are protected by locks so that they cannot be altered by any unauthorized person.

This type of machine has many advantages. It can be installed in the production machine line to give uninterrupted flow production. The total floor space required is in the order of 40 sq ft. By comparison with normal furnace practice this is small in relation to the output capacity of 100 shafts 20 in long by $\frac{3}{4}$ in. diameter per hour. In general, this form of shaft hardening causes much less distortion than other methods, and as a consequence final finishing operations are simplified. In addition, the equipment is particularly suited for selective hardening on such components as rocker shafts and gearbox selector shafts.

Bearing Service

BRITISH Timken Limited, Duston, Northampton, have recently issued a new service manual dealing with the care and maintenance of Timken bearings in the automobile industry. Concise information is given concerning fitting, adjustment and the lubrication of bearings.

Typical sectional arrangement drawings are reproduced showing front wheel bearings, front hub pivots, rear axles, steering gear, final drive, pinion and differential bearings and worm shaft bearings. There are comprehensive notes on the machining of bearing seatings and general practice for cup and cone fitting. There are also brief notes on oil seals and the care and maintenance of tapered roller bearings. (1938)

THE FERGUSON DIESEL ENGINE

A New Power Unit for the TE-F Tractor

THE good pulling characteristics of the diesel engine at low as well as high r.p.m., as evidenced by a flat torque curve, make it a highly desirable power unit for an agricultural tractor. Apart from this, the manufacturers state that the specific fuel consumption of the Ferguson diesel tractor in the field shows a reduction of 30-40 per cent when compared with that of their petrol-engined tractor. This economy is of major significance in countries where there is a substantial price differential in favour of the diesel fuel. The weight of the diesel-engined tractor is 2,786 lb, which is 224 lb. more than that of the similar tractor equipped with a petrol engine developing almost the same power. The good balance and a smooth delivery of torque which are characteristic of this diesel power unit are due, it is claimed, to sound mechanical design and the use of the Freeman-Sanders combustion chamber system.

Combustion chamber theory

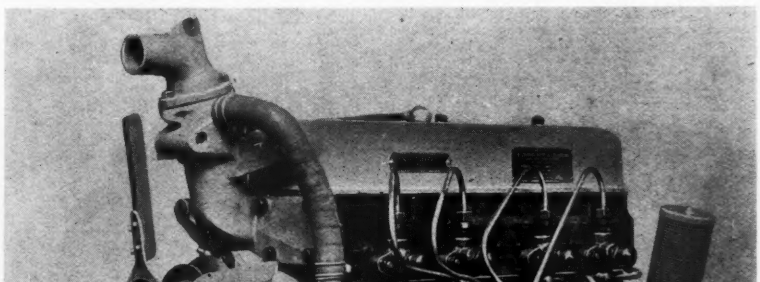
In this system fuel is injected by a pintle-type C.A.V. atomizer, mounted horizontally with its axis $\frac{5}{32}$ in above the centre of the spherical combustion chamber which is offset laterally $1\frac{19}{32}$ in from the cylinder axis. Situated in the cylinder head, this chamber is $1\frac{7}{32}$ in diameter, and the $\frac{39}{64}$ in diameter orifice is flush with the face of the cylinder head. The air cell, machined in the cylinder block, is crescent-shaped in plan and has a depth of $\frac{19}{32}$ in. Its larger radius is formed by the working cylinder bore while the smaller or outer, radius, struck from a centre on the vertical axis of the chamber, is the same as that of the combustion chamber.

The fundamental principle is based upon that of the original Fowler-Sanders arrangement, described in the June, 1937 issue of *Automobile Engineer*. More recent experience, however, has shown that advantages are derived from locating the cell under the combustion chamber, in such a position that combustion is completed as the burning gas leaving the chamber mixes with the air in the cell. Further experiments have proved that the combustion chamber orifice should be circular, as this gives better results over the speed range than the original D-shaped port.

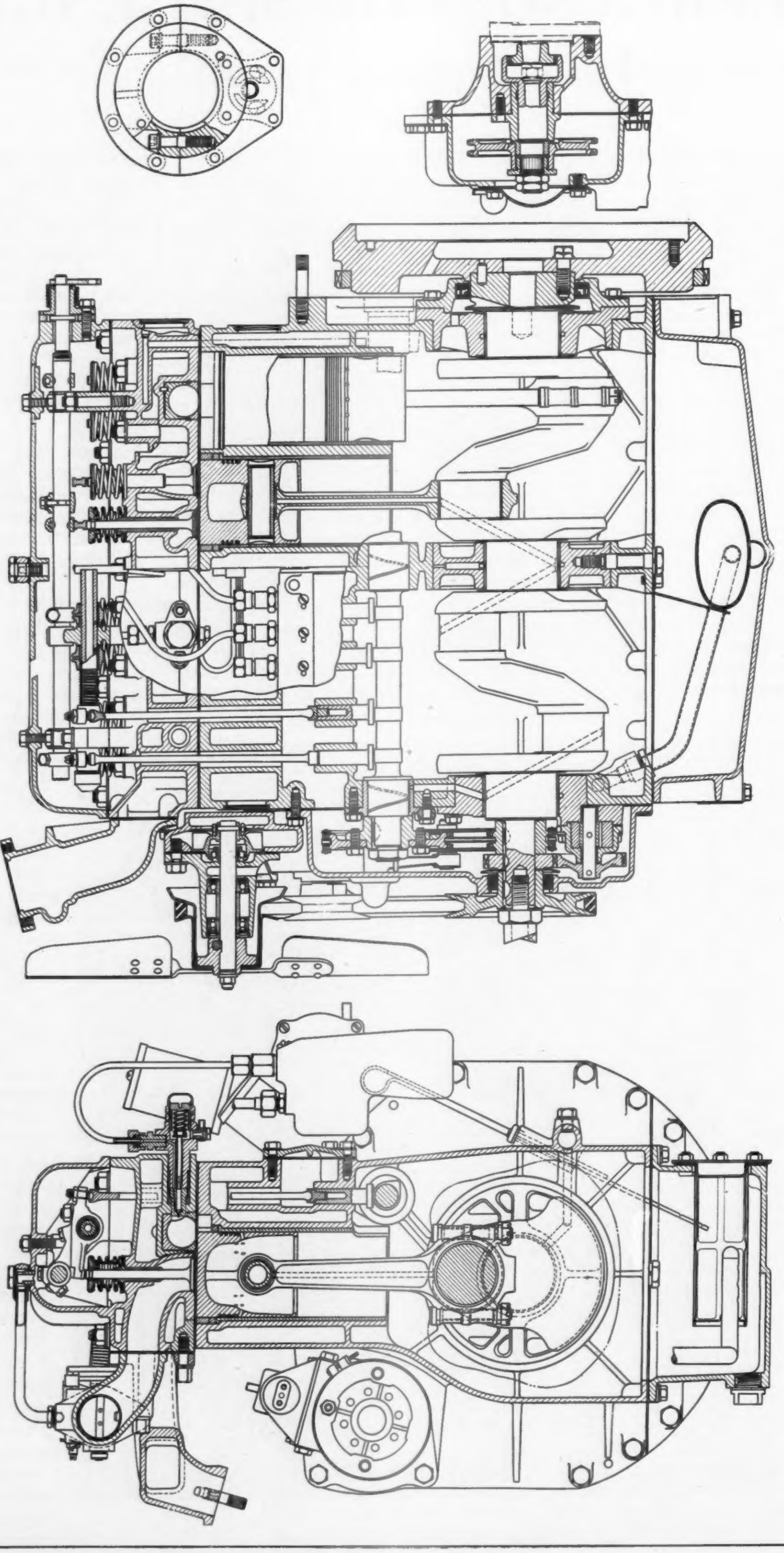
Certain desirable features are still maintained, such as the use of a ductless port which for all practical purposes has no length-of-passage effect. Swirl in the combustion chamber is produced by the squish effect at the end of the piston stroke and the general contour of the combustion chamber, rather than by a long tangential passage. Heat loss is reduced with this arrangement, and a lower specific rate of fuel consumption is obtained. This is demonstrated practically by the fact that even in this relatively small engine, the fuel consumption curve lies about half-way between those of power units of the direct-injection and swirl-chamber types.

It would appear that the circular transfer port between the working cylinder and the combustion chamber has some advantageous effect on the distribution of the turbulent air. In addition, the cross-sectional area of the port is proportioned to promote the turbulence required to match that obtained from a pintle-type injection nozzle. It will be noted that the main features of the first Fowler-Sanders cylinder head are incorporated, namely:

1. Fuel injection occurs during the period of turbulence in the combustion chamber in which there is no uniform rotary swirl. Consequently, fine particles of the spray are not flung against the air cell wall by centrifugal force, and initial combustion is improved.
2. The transfer port not only has no appreciable length, but is of considerably greater diameter than would be the tangential passage of a conventional swirl chamber. These features are made possible by the disposition of the various parts with respect to one another, and the utilization of the squish effect of the piston.



Ferguson four-cylinder diesel engine



FERGUSON DIESEL ENGINE

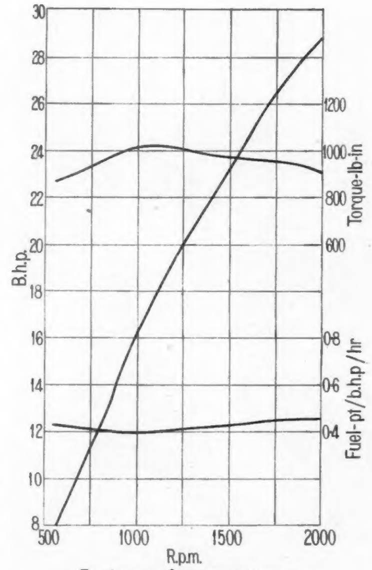
Bore and stroke $3\frac{3}{16}$ in $\times$ 4 in. Swept volume 2092 c.c.
Details: Front main bearing housing and injector drive

A point about which little was known at the time of the development of the earlier chamber was the optimum position of the injector in relation to the hot portion of the cylinder head. Trouble has been experienced in some engine designs due to overheated injectors. This is usually caused by mounting the injector in a position where it is exposed continually to burning gas. In the Freeman-Sanders system, it is claimed, gas in that condition remains at the side of the chamber opposite to the injector which, therefore, does not suffer from overheating.

Principal dimensions and general arrangement

At 2,000 r.p.m. the diesel engine develops 25 belt h.p. at the power take-off pulley. On the test bench with the fan, water-pump, and dynamo fitted, 28 brake h.p. is developed. The engine, a four-cylinder unit, has a swept volume of 2,092 c.c., with a bore and stroke of $3\frac{3}{16}$ in $\times$ 4 in (80·96 mm $\times$ 101 mm). From the power curves it can be seen that the maximum b.m.e.p. and torque are 100 lb/sq in and 1,010 lb-in respectively at 1,050 r.p.m., at which speed the brake specific fuel consumption is 0·402 pt/b.h.p./hr. Maximum b.h.p. is 28·8 at 2,150 r.p.m. The compression ration is 17 to 1.

The layout of the engine departs from conventional lines to provide adequate strength for a three-bearing crankshaft which is carried in barrel type housings in the integral cylinder block and crankcase casting. On the left-hand side is a three-bearing camshaft. Four bearings in pedestals on the head support the rocker and decompressor spindles. On the left side of the engine all the injection equipment is mounted, and on the right side are the manifolds, dynamo and electric starter. At the front a triangulated V-belt arrangement is used to



Engine performance curves

drive the dynamo, fan and water pump from the crankshaft. Owing to the current supply position, material specifications are in many cases substitutes for those normally used, and are liable to further change.

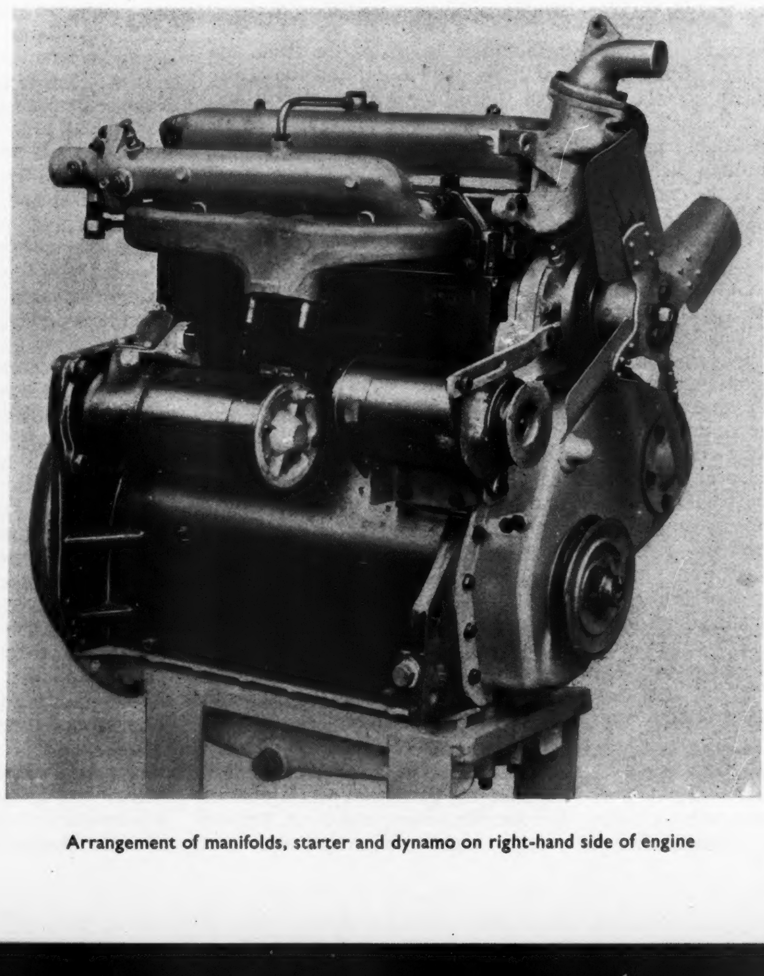
Cylinder block, crankcase, and liners

Made of BS 1452/17 grey cast iron, the integral cylinder block and crankcase incorporates the usual cooling arrangements. Formed in the side of each cylinder bore at the top is the air cell. A die-cast aluminium N.F. 17 sump is fitted, the lower face of the crankcase being 5 in below the centre line of the crankshaft. Bolts in the rear face of the crankcase and sump carry the clutch housing.

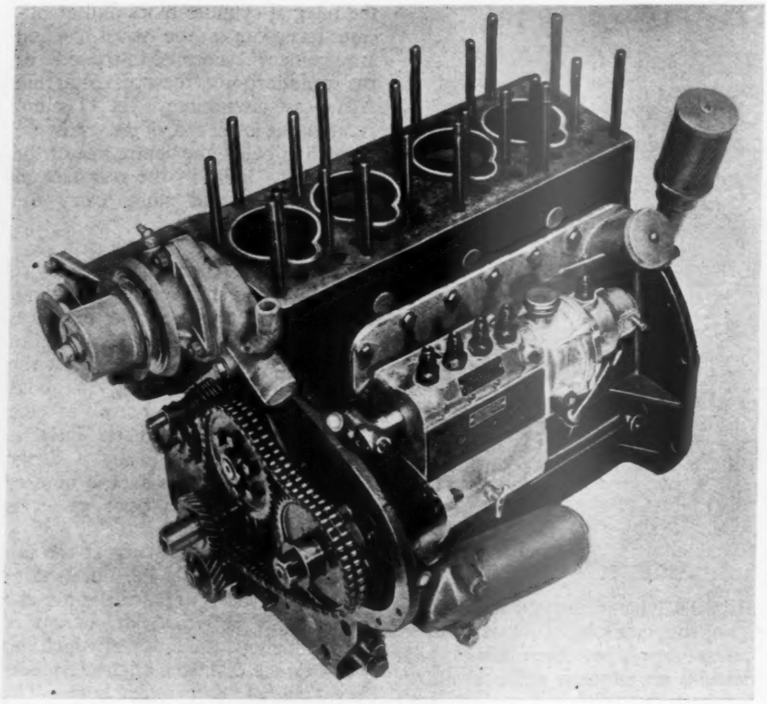
The slip-fit, centrifugally-cast, dry-type cylinder liners are made of Brivadium and supplied by the British Piston Ring Co. Their wall thickness is $\frac{7}{64}$ in, and that of the casting in which they are housed is $\frac{7}{32}$ in at the wetted surface, increasing to about $\frac{13}{32}$ in between the bores of the adjoining cylinders. Each liner is located at its top end by a flange seating on a shoulder in the bore of the block. A thin copper gasket is fitted under the flange.

A patented arrangement at the top of the liner controls the build-up of carbon on the piston land above the top compression ring. It consists of two cuff rings, the lower of which has an internal diameter slightly smaller than that of the cylinder bore. The upper ring, made of 4K6 chrome-iron No. 1, has a gap $1\frac{1}{4}$ in wide machined in it to form part of the air cell. The two edges of the gap are equally spaced from and parallel to a plane containing the axis of the cylinder. A conical seating on the upper face of the chrome-iron 4K6 No. 5 lower ring prevents the gap from closing when the ring is in position in the cylinder. Had a single ring been used instead of the two rings, the change of section at the cut-out could have been a potential source of weakness.

A cut-out machined in the base of the liner coincides with a similar gap cast in the skirt in the crankcase to facilitate gudgeon pin removal. Three bosses in the crankcase constitute the camshaft bearings, and the tappets are accom-



Arrangement of manifolds, starter and dynamo on right-hand side of engine



Cylinder head removed to show air cells

modated vertically in drilled bosses in the usual manner.

The general layout of the engine provides for assembly of the crankshaft and flywheel unit with its centre and rear main bearings, from the opening at the rear of the crankcase. For this purpose the crankshaft bearings are mounted in large-diameter, split carriers; the through diameters at the centre and rear of the crankcase being $6\frac{3}{4}$ in to clear the crank webs. The forward carrier is 5 in diameter only, as the clearance over the webs is not required. Die-cast in N.F. 21 magnesium alloy, the carriers are split horizontally and held together by two En 24 socket bolts locked by spring washers. The bolts pass through En 18 dowel tubes which provide lateral location. The centre bearing carrier is secured in the crankcase by a bolt and spring washer at the centre of the lower edge of the crankcase web. At the rear, the carrier is flanged and bolted to the crankcase wall, together with the BS 1452/10 iron casting which houses the lip-type Super Oil seal for the crankshaft. The front bearing housing is secured by set-screws to the front crankcase face. Substantial ribbing in the crankcase walls, and webs, and bearing housings is provided.

Crankshaft, connecting rod and piston assembly

Forged in En 110 manganese-molybdenum steel, the crankshaft is statically and dynamically balanced. Integral balance weights are formed on the webs at each of the three bearings. All journals are $2\frac{3}{4}$ in diameter. The length of the front and rear bearings is $1\frac{1}{2}$ in, and the centre $1\frac{5}{8}$ in. Lead-bronze precision bearings, lead-indium plated, are employed for the main journals and the big ends, which are $2\frac{15}{16}$ in diameter and $1\frac{7}{32}$ in long. End location of the crankshaft is effected by split thrust rings of the same material as the bearing shells, one on each side of the rear journal-bearing. The rear extension of the crankshaft, incorporating an oil thrower ring, has a ground face that bears against the thrust ring. Located by a dowel, the BS 1452/17 cast-iron flywheel carrying the flame-hardened En 8 D starter ring is bolted to the 4 in diameter rear end of the crankshaft which is counterbored to carry a ball bearing for the clutch shaft.

Mounted at the front of the crankshaft are the En 8 R sprocket for the two-row roller timing chain, the BS 1452/17 oil pump drive gear, a pressed-steel oil-thrower ring, and the V-belt pulley. Three Woodruff keys transmit the drive, and the assembly is pulled up by the usual nut incorporating the starting dogs.

In the N.F. 17 aluminium die-cast front cover there is a Super oil seal. A projecting lip on the cover, shrouded by the fan pulley, assists in keeping foreign matter from the seal, and further protection is provided by a felt washer incorporated in the seal. Tension in the timing chain is maintained by a jockey sprocket mounted on a spring-loaded arm. One end of the arm is carried in the cover, the other end bearing in the crankcase wall.

Drilled for the passage of oil from the big-end to the small-end, the I-section connecting rods are forged in En 16 manganese-molybdenum steel and have a centre-to-centre length of 8 in. The big-end bearings, split at right angles to the centre line of the rods, are secured by two En 111 low nickel-chrome steel bolts with slotted nuts. Vandervell Clevite 10 small-end bushes are fitted. The gudgeon pins of En 33 3 per cent nickel-chrome steel are 1 in outside diameter and $\frac{5}{8}$ in inside diameter, and are located by circlips. The flat-topped pistons are in Wellworthy 52 aluminium alloy, anodised in the ring grooves only. Three compression rings, the uppermost being chromium-plated, are 0·078 in wide and 0·125 in thick. Drilled holes, both in and below the ring groove, carry oil away from the slotted scraper ring, which has a face width of 0·156 in and a thickness of 0·117 in. All rings are carried above the gudgeon pin, the top one being about 0·875 in below the piston crown.

Camshaft and valve gear

The Monikrom shaft has bearings $1\frac{9}{16}$ in diameter. At the front and rear ends they are $1\frac{3}{8}$ in long; the length of the centre being $1\frac{3}{4}$ in. A cap seals off the rear bearing from the clutch housing. On a flanged hub keyed to the 1 in diameter forward extension of the camshaft, is mounted the En 8 R timing sprocket. The flange has six unequally pitched $\frac{5}{16}$ in tapped holes while the sprocket has equally spaced holes to furnish a means of fine adjustment of timing. One hole in the sprocket gives positive location and the other five holes are elongated.

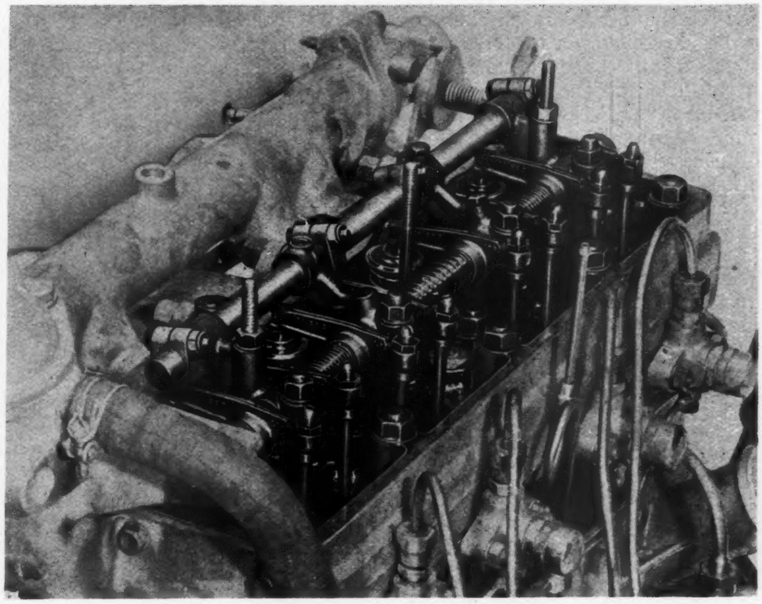
The Bricast BS 1452/17 tappets are fitted with hardened En 36 nickel-chrome steel cup-shaped inserts. The push rods, $\frac{1}{4}$ in diameter, $8\frac{1}{2}$ in long, are of En 8 steel. They pass through passages cast in the cylinder block and the head. A ball-end at the base seats in the tappet insert, and at the top a cup receives the spherical end of the case-hardened En 32 B adjusting screw fitted in the rocker. One extension of the die-cast aluminium tappet cover forms the oil filler, and another carries the Vokes filter for crankcase breathing.

The case-hardened En 32 B hollow rocker spindle has two end caps, pegged in position, to locate the end rockers and to seal the spindle bore. It is mounted in four BS 1452/14 grey cast-iron bearing pedestals. Location

of the spindle is by a bolt screwed through the third pedestal. Two Vandervell phosphor-bronze bushes are pressed into the En 32 B rockers, one from each side, so that there is an annular oil channel between them. Pads on the ends of the rockers are case-hardened where they contact the decompressor cams and the valve stems. Lateral location is by the usual helical springs.

Extensions of the pedestals carry the En 8 decompressor spindle with its axis above the valve stems. End location is effected by the dowel-end of a bolt engaged in an annular groove in the spindle at the third bearing, thus allowing it freedom to rotate. The decompressor cams are clamped in position by a pinch bolt which locates in a groove around the spindle. When in the released position there is a clearance of 0·03 in on three cams and of 0·04 in on the fourth. This provides first for three-quarter decompression and then full decompression as the lever is moved to engage in appropriate notches in the quadrant on the control panel. Sealing is effected by a felt ring around the short control spindle and located between the casing and the lever welded on the outer end of the spindle. A positive stop is provided by a pin that passes through the centre of the spindle and operates in an enclosed quadrant plate. At the inner end, a tongue and slot arrangement connects the control to the decompressor spindle. A coil return-spring is fitted on the control spindle casing which, together with the stop quadrant, is bolted to the N.F.17 aluminium diecast rocker cover.

Both inlet and exhaust valves have a stem diameter of 0·3125 in; the diameter of the inlet valve head being $1\frac{1}{4}$ in, while that of the exhaust valve is $1\frac{1}{8}$ in. Tappet clearance for both valves is 0·012 in when cold and the valve lift is 0·3075 in inlet, and 0·342 in exhaust. The timing overlap is 10 deg; the inlet valve opening 5 deg before T.D.C. and closing 25 deg after B.D.C., and the exhaust opening 45 deg before B.D.C. and closing 5 deg after T.D.C. Silichrome steel is used for the inlet valves whilst the exhaust valves are in XB valve steel. The BS 1452/14 grey cast-iron valve guides are not interchangeable as their lengths are 1·875 in and 2·375 in for the inlet and exhaust valves respectively. This arrangement provides a large area of metal to conduct heat away from the exhaust valve and on the inlet side it leaves the passage as free from obstruction as possible. A collar, cast near the top of the guide, seats on the cylinder head and carries a pressed-steel washer on which the two concentric valve springs bear.



Rocker and decompressor gear

Split collets on top of the valve stems secure the springs in the usual manner.

The BS 1452/14 grey cast-iron cylinder head has ample cooling space, particularly around the valve ports and guides and the combustion chamber. Twenty-two $\frac{7}{16}$ in diameter En 111 studs are used to hold the head down and the usual copper-asbestos gasket makes the cylinder head joint. Three studs are provided to hold down the rocker cover, a Corflex washer being used at the face joint. A union is fitted to the top of the cover for a pipe connecting it with the inlet manifold. Bolted to the front of the head is an N.F. 24 aluminium thermostat housing.

The spherical combustion chamber is machined in a mild steel block of cylindrical form, which is split into an upper and lower part. The upper part, held against rotation by a dowel, is secured in position by the lower part which is screwed into its housing in the cylinder head by means of lugs on its lower face. These lugs are subsequently machined off. The cylinder top cuff ring and gasket bear against this part and prevent it unscrewing in service.

Lubrication system

Running at crankshaft speed the drive spindle of the Hobourn-Eaton oil pump is supported in two Clevite 10 plain bearings. One of these is mounted in the magnesium alloy main journal bearing housing and the other in the cast-iron oil-pump body. Mills pins secure both the En 8 R crankshaft-driven gear and the pump rotor to the spindle. Axial location is effected by the pump rotor in its casing.

Oil is drawn from the sump, which has a capacity of 12 pints, through a gauze filter and a steel tube of 0·532 in inside diameter. The pump then passes it to the relief valve, which maintains the pressure at 40 to 60 lb/sq in, and through a Purolator Micronic filter to the oil gallery drilled in the side of the crankcase. At the rear end of the gallery a union is fitted for a pipe connection to a pressure gauge. From the gallery, drilled ducts in the crankcase and bearing housings convey the oil to an annular groove around each journal bearing. Ducts from these grooves are provided for the lubrication of the camshaft bearings, and drillings in the crankshaft pass oil to the big ends whence it flows up the drilled connecting rods. Oil is piped from a union, connecting with the front camshaft lubricating duct, to provide a pressure jet of oil over the timing chain jockey sprocket, the oil pump drive gear, and crankshaft sprocket. At the rear end, a vertical drilling from the camshaft bearing supplies oil through the rear bearing pedestal to the hollow rocker spindle. Oil passes to the rocker bearings through small holes drilled in the spindle. A hole in each rocker passes oil into a groove on the top surface which carries it along to lubricate the pushrod end only. The tappets are lubricated by splash from the crankcase and by oil running down the pushrods.

A water pump of the impeller type is bolted to the front of the cylinder block and driven by a V-belt from the crankshaft; the triangulated drive including the dynamo as is usual. The drive spindle is mounted on two

ball-bearings carried in an extension of the pump body. On the forward end of the spindle is the welded pressed-steel, driving pulley together with the four-bladed fan bolted to a stamped hub. A cotter pin in the stamping furnishes the drive. Spaced by a 1 in distance piece, the bearings are lubricated by means of a grease nipple on the pump body. Between the impeller assembly with its spring-loaded, rubber gland incorporating a moulded-in graphite bearing face, there is a draining space and thrower to prevent any water leak from reaching the ball bearings.

Fuel injection and starting systems

C.A.V. fuel injection equipment is fitted, which embodies a BEP/MN80A/102X variable speed pneumatic governor. This is mounted on the rear end of the BPE/4A6OQ/120S/6200EL injection pump. The usual enrichment device is incorporated for starting, but it is automatically returned to normal as the engine picks up. Type BDN 431 fuel injector nozzles are used, and the nozzle holders are type BKB/50S/622. Injection cut-off is timed to occur 30 deg before T.D.C; the working pressure being 1,500 lb/sq in. Fuel is supplied to the injection pumps through two Volkes filters in tandem.

An En 3B mild steel locating bush carries the injection pump drive sprocket. Splined into a forward-projecting boss on the sprocket is an En 8 R driving sleeve which is also internally splined on to the spindle made of En 30 A air-hardening, nickel-chrome, leaded steel. Two nuts on the front end of the spindle pull the whole driving wheel assembly up against a shoulder and lock it, whilst at the same time clamping the wheel between the flanged ends of the locating bush and the drive sleeve respectively. The flanged BS 1452/14 grey cast-iron bush for the spindle is secured to the crankcase by three bolts. The rear end of the spindle is counterbored to carry the toothed injection pump driving gear. A domed, pressed-steel plate bolted on to the front cover can be removed to provide access for timing adjustment.

A BS 1452/10 cast-iron exhaust manifold and an N.F. 17 die-cast aluminium manifold are fitted. At its rear end there is a union provided for the pipe connecting it to the pneumatic governor. Air filtration is effected by an A.C. centrifuge-type filter in conjunction with a special A.C. oil bath filter with a detachable base.

For cold starting, a Kigass hand-operated priming pump forces starting fuel on to an electrically heated plug through an atomizer nozzle in the inlet pipe. The fuel is ignited and drawn into the engine cylinders, so warming the induction system and combustion chambers. Starting is effected by a Lucas pre-engaging type starter. Two Lucas T/TX19/TE six-volt batteries in series supply the 12-volt system. They have a capacity of 115 ampere-hours at a 10-hour discharge rate. Driven together with the fan and water pump by V-belt, is a Lucas 12-volt shunt-wound dynamo. A separate compensated voltage control regulator is employed.

CORRESPONDENCE

Correspondence on subjects of technical interest is invited. The name and address of the writer must be given, though not necessarily for publication. No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter. If a reply by post be desired, a stamped addressed envelope should be enclosed.

THE GREASE GUN

SIR,—Your Editorial in the October issue strikes at a matter of considerable importance to motor manufacturers, makers of lubricating equipment, oil companies, garages, and motor car users. It is amazing that after fifty years or more of motoring, we still rely on an old method long since superseded on many industrial machines. To expect the modern motorist to get underneath his car with a grease gun almost passes comprehension. The modern driver is supplied with quick-lift engine bonnets, self-cancelling direction indicators, easy-clean wheels, and other labour-saving refinements. He has self-changing gear boxes, automatic overdrives and many details which tend to make his motoring smooth and trouble-free. You do well to arouse the manufacturers.

Your readers should not be put off, however, by your reference to automatic lubrication systems or the lubricants for use therein. It should be remembered that such systems are available for commercial vehicles, that many makers fit them as standard, that they have been adopted by London Transport, and all their trolley buses are fitted with them. Wear on steering pins, after almost a life-time's work, has frequently been reduced to nothing. Certainly they cost money, but they save money in maintenance.

These systems could be fitted to motor cars if the manufacturers wished. It must be remembered that Rolls-Royce has employed such a system for twenty years, and they are fitted to Rover, Bentley and Daimler cars also. If a good system were to be developed for popular cars, it could be manufactured in such large numbers that the cost would not be great. The position at present is that the motor manufacturers have a sellers' market and they do not need to bother about these things.

Other reasons why nothing has been done may be that:—(1) a really low-cost, foolproof system has not been developed for small cars. (2) The garage industry is against it, as also are the makers of lubricating equipment. Point (1) does not present any obstacles. Sympathy would be with garage proprietors and equipment manufacturers, since many people would not visit a garage repair shop if their vehicles never wanted greasing.

Lubricants being made today will function satisfactorily under far worse conditions than those encountered by the steering and suspension joints of road vehicles. In industrial applications, lubrication requirements are frequently much more exacting, yet no particular difficulty is experienced in meeting demands with either grease or oil.

Regarding your remarks concerning grease guns, most people will again be in complete agreement. Recharging grease guns is a job which no motor manufacturer should expect his customers to perform. At the Motor Show this year, there will be an answer. This is a gun that has recharging cartridges, sold wrapped in paper. All that one does is to strip off the paper and insert the grease in the gun in one operation. Providing the grease supplied in this way does everything that a chassis grease should, and the maker's say that it does, this is certainly progress.

Yours faithfully,

E. V. PATERSON,
(Editor, *Scientific Lubrication*)

SHOW REVIEW NUMBER

The extra issue of the "Automobile Engineer" will be published as usual in connection with the London Motor Show. It will constitute a critical review of the more interesting exhibits, including coachwork, and will have numerous illustrations of special features and design characteristics.

The Show issue will be available on November 15th, and can be obtained by order from newsagents throughout the United Kingdom, price 3s. 6d. net. Readers are reminded that it is still necessary to make arrangements with the newsagent to ensure that a copy is secured.

PRECISION MEASUREMENT

An Exhibition of Modern Measuring Equipment

PRECISION manufacture and precision measurement are indispensable to that ready mating and interchangeability of parts upon which efficient and inexpensive quantity production is based. Since quantity production of so complex a finished article as the motor car or commercial vehicle involves huge numbers of components, the importance of measurement methods that shall be simple and quick, as well as accurate and reliable, needs no emphasis. An exhibition of precision measuring equipment was staged, during October, in the Vauxhall Bridge Road showrooms of Alfred Herbert Ltd., machine-tool manufacturers.

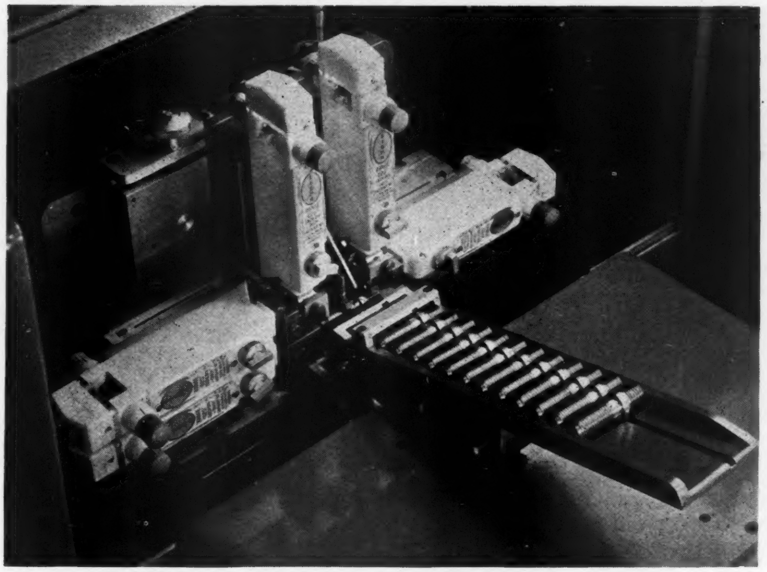


Fig. 1. Magazine feed to Sigma multi-dimensional inspection machine

Developments in machine tools and production methods have tended to outpace developments in methods of inspection, with the result that percentage checking has been resorted to as a means of circumventing inspection bottle-necks or inordinate non-productive wage-bills. Quality control has not been an unqualified success: its principles are not always appreciated by the operators upon whom the system depends for its effectiveness.

Multi-dimensional inspection machines

Sigma automatic, multi-dimensional inspection machines, designed and developed by the Sigma Instrument Co. Ltd., Letchworth, permit rapid 100 per cent inspection, to an accuracy of 0·00005 in, of thousands of parts per hour. One man can operate four of these machines fitted with hopper feed, each machine having an output of up to 4,000 parts per hour. When magazine feeds are used, as illustrated in Fig. 1, outputs are limited to about 2,000 per hour, by the speed at which the operator can feed the parts. In both cases a workholder mounted on a reciprocating slide transfers the parts to the gauging position.

Standard fully automatic machines can accommodate parts of up to 2 in diameter and 8 in length, and check as many as 12 dimensions simultaneously. From the standard and interchangeable units, combinations of electrical gauges, signal units, workholders and hoppers appropriate to particular requirements are supplied. Provision is made for automatic delivery and sorting into drawers or boxes containing accepted, rejected, or rectifiable parts, and the result of each inspection is indicated by signal lamps. Counters may be incorporated in the machines. Each of the six sizes of machines is supplied in three forms for checking (1) shafts, and spindles, (2) complex parts having profiles, depth and internal diameters, and (3) different components. In the third or Simplex form, all gauges are mounted on a removable bracket, so that another assembly can be substituted in about 10 minutes.

Fig. 2. Hilger Universal projector

Similar, semi-automatic, multi-dimensional inspection machines, intended for more delicate or larger components of up to 4 in diameter and 8 in length, depend upon hand loading, unloading, and grading according to signal indication. As many as 1,500 components may be dealt with per hour. The operator places the component on or in a fixture carried upon a reciprocating slide feeding the inspection point, and the movement of the slide is controlled by switches. Hand-operated machines are available for components inspected in smaller quantities (a few hundreds per hour) and of up to 2 in diameter and 8 in length. Feeding, withdrawal, and sorting of the parts are wholly by hand but the gauging is done, and signal indications are given, automatically. Change-over from one type of part to another can be effected in about 10 minutes, the gauges being mounted on compound slides to facilitate adjustments.

Projectors

Several projectors developed by Hilger and Watts Ltd., in collaboration with Alfred Herbert Ltd., were exhibited. Of these the Hilger Universal projector, Fig. 2, is of general utility, meeting average toolroom and inspection department requirements. This model provides for comparison with enlarged layouts of magnified images of form tools and profiles, and, in addition, for co-ordinate measurement to 0·0001 in by micrometer equipment, and angular measurement to one minute by precision protractor. The amount of an error can then be determined by the

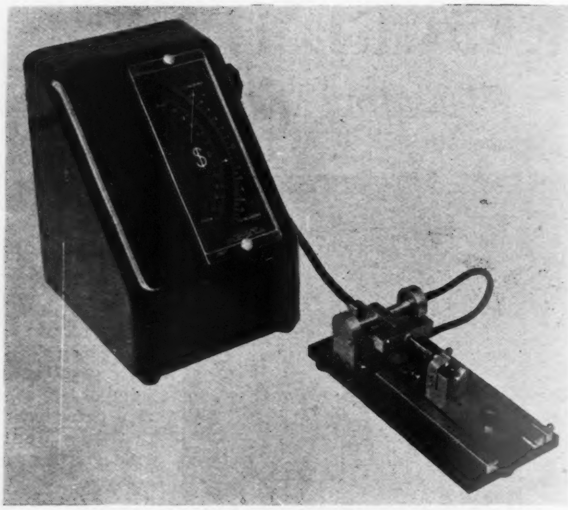


Fig. 3. Sigma pneumatic comparator

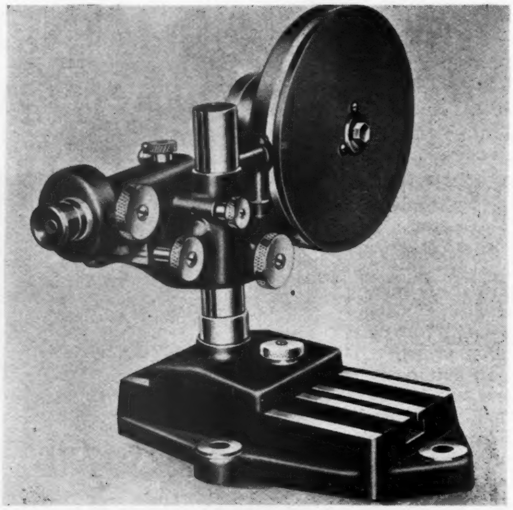


Fig. 4. Watts circular-division tester

movement of the workpiece required to align any portion of its projected image with the corresponding portion of the standard representation on the screen. The ability to dispense with gauges is more especially of value in the inspection of pieces, such as sparking-plug insulators, of which the abrasive material greatly reduces the gauge-life. Maximum dimensions of the work range from 6 in diameter and 8 in length, between work-table centres, at $\times 10$ magnification, to 1 in diameter and 8 in length at $\times 100$ magnification, the projection lenses being supplied for $\times 10$, 15, 25, 50 and 100 magnifications.

The large Universal projector takes 6 in $\times$ 12 in components at low magnifications, and 2 in $\times$ 12 in components at $\times 100$ magnification. All models can be equipped for surface projection, to deal with obscured forms, as well as for silhouette projection. Inspection speeds are high by normal standards. On a Hilger Production projector, for example, sparking-plug insulators can be compared at a rate of 750 per hour, with a template giving all permissible tolerances. This machine does not provide for the measurement of errors by adjustment.

Comparators and other equipment

Among the Sigma mechanical or electrical comparators exhibited, was a vertical, mechanical comparator with a ten-station, indexing turret-attachment. A fixture for the inspection of a single dimension is provided at each station. The adjustable contact tip of each fixture is set so that a perfect component will give a zero reading on the indicator, and, to avoid confusion, the tolerance boundaries for the various dimensions are marked on the dial of the indicator in different colours corresponding to the colours carried by the turret-stations for these dimensions. A simpler device by which one comparator may be used for inspecting several dimensions is the provision of a number of fixtures of such size that fixture-height plus component-dimension is a constant, giving zero reading for a perfect component. The vertical mechanical comparators have vertical capacities of from 6 in to 24 in and magnifications of 500, 1,000 and 3,000. Graduations are in 0.0001 in and 0.005 mm at magnification $\times 500$, and in 0.00002 in and 0.0005 mm at magnification $\times 3,000$. The electrical comparator is supplied with magnifications from 10,000 to 50,000; both coarse and fine adjustments are provided by knurled screws, the desired setting being obtained by the use of slip gauges. The stylus can be lifted while the component for inspection is being inserted.

Sigma pneumatic comparators, of which one is shown in Fig. 3, are intended particularly for high-precision measurement of small internal diameters. Air supplied at 50-100 lb/sq in by a small orifice, escaping through the gap between the component inspected and a mandrel upon which it is placed, provides, in its quantity, a measure of the difference between component and mandrel dimensions. Operation of the pointer over a scale of magnification $\times 5,000$, involves application of the Wheatstone-bridge principle to a resistance varied by the play of the airstream upon it. Accurate measurements may be made on completion of a machining operation, even though a film of coolant remains upon the component. Concentricity of a diameter can be tested by rotating a component upon the mandrel, and noting the movement of the gauge pointer.

The Watts circular-division tester, shown in Fig. 4, is for checking dividing mechanisms, and has a master circle graduated at angular intervals of 10 minutes, together with a microscopic unit sub-dividing these graduations into intervals of 6 seconds. The Cooke optical dividing head is a high precision instrument providing a reference standard for circular division and a head robust enough for finish machining operations in the toolroom. The scale reads to 0.5 minutes and the operator can estimate to 6 seconds. Other exhibits included a pitch comparator and an effective diameter measuring machine for screw threads; a tool dynamometer for the measurement of axial, radial and vertical pressure upon tools, and surface-flatness testing equipment.

Alfred Herbert Ltd., Coventry, are the sole agents in the British Isles for Sigma multi-dimensional inspection machines and comparators, and sole retailers of Hilger projectors.

Bronze Castings

David Brown & Sons (Huddersfield) Ltd., Meltham, near Huddersfield, have issued an 83 page publication for users of bronze castings. This booklet, which is entitled *Taurus Bronze Castings*, includes in its contents "The Art of Bronze Founding", "The Selection of Bronze for Engineering Castings" and "The Influence of Composition on Physical Properties" as well as many other sections on engineering practice as applied to the use of bronze. It is particularly well illustrated and one of its outstanding features is a series of coloured reproductions of photographs of the microstructure of various bronze alloys.

David Brown & Sons state that they have endeavoured to outline in a way suited to the needs of the engineer rather than the metallurgist, the more important considerations that enter into the selection of bronzes, such as material and analysis, methods of casting and the necessary physical properties to meet particular conditions of service. Specifications are given of twenty-one Taurus bronzes, some of them adapted to both sand and centrifugal casting.

NOISE REDUCTION

Part 1: Consideration of Vehicle Design Factors

THERE is no single, simple measure that by itself can make a noisy car quiet. The subject of noise reduction is a very complex one, and merits careful and detailed study. This article aims to outline the measures to be taken, in both the design and development stages, to ensure that the best possible results are obtained. Emphasis, therefore, is placed on the practical aspects of the problem, and the reader is advised to refer to a text-book on the theory of sound if he requires more detailed information on the fundamental principles involved.

Units of noise measurement

Considerable confusion is often experienced in connection with the units used in sound measurement. The simplest description of these units is that the decibel is a logarithmic unit of measurement of the pressure generated by the sound waves in air. In other words, it is a precise, scientific measurement of a physical phenomenon. On the other hand, the phon is a measure of the actual loudness of sound as experienced by a normal human ear. Fig. 1 was obtained by careful experiments with hundreds of male and female subjects, and illustrates how sound pressure level and loudness are inter-related. It will be observed that the sensitivity of the human ear varies considerably over the audible frequency range. For example, at 1000 c/s the sound pressure level in decibels has by definition the same numerical value as the loudness in phons, but to obtain the same loudness as 40 decibels at 1000 c/s, it is necessary at 100 c/s to increase the sound pressure level to about 60 decibels. A noise level of 40 phons corresponds approximately with that of a very quiet room in the average home.

One decibel is defined as the smallest sound which can be detected by the average human ear. At the other extreme, the threshold of feeling is reached at about 120 decibels. This is the point where the sensation experienced by the ear changes from one of noise to a sensation of feeling or pain.

From the logarithmic nature of the decibel unit it follows that 20 decibels, for example, is very considerably more than twice the sound pressure level of 10 decibels. The law of addition of decibel units is shown in Fig. 2. Referring to this diagram it can be seen that if two sounds of unequal pressure level are heard simultaneously, then a decibel meter reading of the combined sound will be somewhat less than three decibels higher than the value of the louder of the two sounds.

The audible frequency range is from about 24 c/s to, say, 20,000 c/s. These figures cannot be stated with precision as the sensitivity of the ear varies from person to person and, indeed, from time to time according to changes in the health and age of the individual. In motor vehicles the frequency range of most interest is from the lowest audible frequency up to about 1000 c/s.

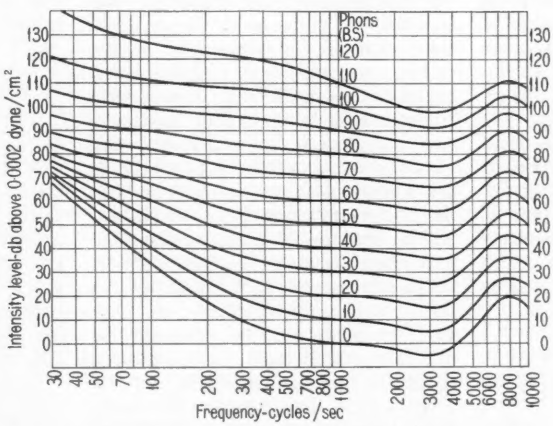


Fig. 1. Relationship between loudness intensity and pressure-level intensity

The frequency spectrum and the psychological aspect

A typical noise spectrum giving the sound pressure level of different frequencies at various road speeds is shown in Fig. 3. The points indicated by the crosses on the diagram represent meter readings obtained in conjunction with an octave filter. This can be set to give sound pressure level readings on one octave of the frequency scale at a time. It is not strictly correct to connect those readings to form a continuous curve, since it is improbable that readings taken between adjacent points would fall on or near the line connecting them. Furthermore, the point readings represent the pressure level of a combination of frequencies, over a range of one octave, which are centred at the frequency indicated. However, this method of presentation has been adopted, so that the general trend of the readings at any given vehicle speed may be easily followed.

Opinions are widely divided on the subject of what constitutes an objectionable noise, and the reader must, perforce, draw his own conclusions in this matter. Various authorities have stated that intermittent or variable pitch noises are annoying, and that high frequency noises are more objectionable than lower frequency ones of the same intensity. It is a fact, however, that noise levels in the region of 75 to 80 phons, if sustained over a period of 10 to 15 years, may cause deafness. for short periods, a level of 80 to 100 phons is tolerable. The fatiguing effect of noise is not as widely appreciated as it should be. It has a particular bearing on road accidents and is most important to road hauliers. A persistent noise, such as the hum of an engine, causes nervous energy to be used up as the mind is continuously trying to disregard it. This process is contrary to natural tendencies. From the psychological point of view, it is natural for the brain to be continuously receptive to noises, as these give warning of danger. It is only necessary to recollect how tired one feels after a long railway journey to realize how much fatigue is caused by noise. There is no question of nervous energy being used up by back-seat driving in a train and thus the train presents a more convincing example of noise fatigue than does a car.

The nature of noise

It is important that the fundamentals of noise reduction should be fully understood and applied by the designer, as frequently a car which

basically is not correct, cannot be made quiet at the development stage without some major redesign. That is usually too expensive a measure to contemplate. The most satisfactory way to keep down the noise level in a vehicle is to reduce or eliminate it at its source. Hence, the first task is to determine accurately the source.

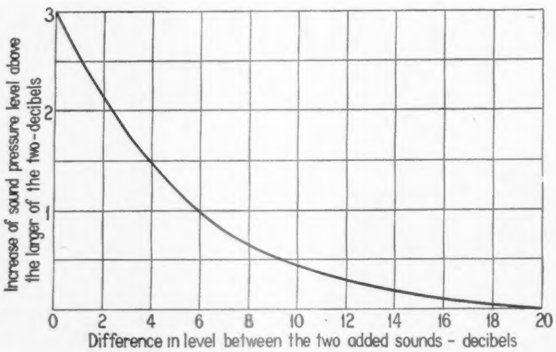


Fig. 2. Law of addition of decibels

Noise is a vibration of the air. In a car, it is generated mainly by vibrating parts of component mechanisms and metal panels. Some of the noise caused by vibration is transmitted through the structure and some is airborne. It is obvious that the greater the area of surface generating the noise, the greater will be the volume of sound produced. This principle may be demonstrated with a tuning fork and sounding board. Most noise troubles in a motor car are due to the amplification, by the large body panels, of vibrations transmitted to them from chassis and engine parts.

Resonance

Excessive amplification of noise is usually due to resonance, the nature of which may here be briefly considered. If a weight suspended from the lower end of a helical spring is pulled down and then released, it will oscillate up and down at the natural frequency of the spring. The frequency remains constant but the amplitude of motion becomes less and less until internal friction in the spring and air resistance finally bring it to rest. It follows that if sufficient energy is supplied in phase with the vibrating system to make good that lost by friction the system would continue to vibrate. If energy is supplied at a slightly higher rate, then the amplitude will progressively become larger. In practice, however, friction increases as the amplitude is extended and eventually the energy input is balanced by the friction losses and the amplitude will be stabilized. Only a very small energy input is necessary to overcome friction when the amplitude of motion is small. Conversely, if the energy is supplied out of phase with the vibration, it will quickly stop the motion.

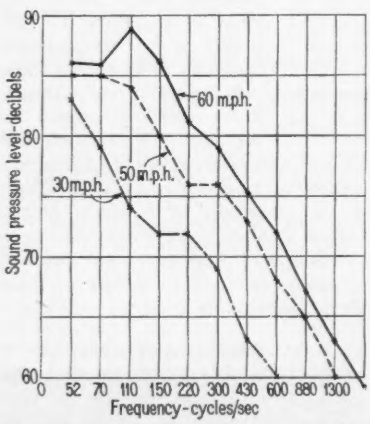


Fig. 3. Frequency spectrum for a typical 2-litre saloon car

The stiffness of a motor body panel provides the spring force, and its weight completes the system necessary for vibration. If energy, in phase with its natural frequency, is supplied to the system by an engine vibration, for example, then the amplitude will increase to perhaps three or four times that of the engine part transmitting the vibration. This is the condition of resonance. Should the frequency of the engine vibration not correspond with that of the natural frequency of the panel, then the energy will not be supplied in the correct phase and the amplitude will be lessened. A resonant vibration may be induced if the exciting frequency is a multiple of the natural frequency. For example, it may supply energy to the vibration on alternate oscillations. Furthermore, there is more than one natural frequency. The lowest of these, termed the fundamental, is likely to be the most important as it is usually of the largest amplitude. The other natural frequencies, or higher harmonics, are multiples of the fundamental frequency.

Isolating the structure from vibration

The elimination of vibration in engines, gearboxes, propeller shafts and other rotating components is a specialized subject and will not be discussed here. Assuming that everything possible in that direction has been done, the first step is to insulate the source of noise from the frame and body. This is most important in chassisless construction and only slightly less critical in the case of the vehicle having a separate chassis frame and body.

The tendency in the development of the motor vehicle structures has been towards lighter chassis frames, with bodies taking more of the load. Frames are of much less depth than the body panels and for that reason are more flexible—assuming, of course, that the body design is structurally sound. This results in the body taking a much larger proportion of the loads than is generally realized. Consequently, it is not sound engineering practice to insulate the body from the frame with rubber mountings. Where this is attempted, the holding-down bolts frequently so compress the rubber that its value as a vibration isolator is negligible. Even when correctly designed and assembled, the result is a compromise, giving a slight reduction in noise level in the car, but at the same time an undesirable reduction in the stiffness and strength of the structure. The more correct procedure, therefore, is to isolate the power unit, steering, suspension and transmission components from the frame by means of rubber.

Engine mounting and suspension isolation

The engine must be supported on its rubber mountings so that it is free to vibrate both vertically and torsionally. The natural frequency of the mounting in both these modes must be sufficiently low to prevent an undue resonant amplitude of vibration at the normal slow-running speeds. At the same time the mounting must be sufficiently strong and stiff to react the maximum torque

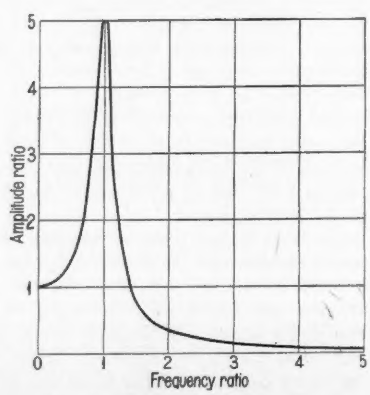


Fig. 4. Typical resonance curve

and the weight of the engine without excessive deflection.

Stresses in the rubber must at all times be below the safe limiting value. The stress limit may be found for any particular rubber mixture by consultation with the manufacturers, but in general it is about 50 lb/sq in or 80 per cent strain, whichever condition is first reached in shear. In compression, the figure is 10 to 15 per cent of strain, which will be found to give a higher stress than in the case of shear. The reason for the tolerance in this figure for compression is that compressive strain depends partly on the ratio of compressed area to the unrestricted side area which is free to bulge. In the case of a thin sandwich the lower figure should be used. Rubber in tension should be avoided as minute surface cracks tend, under this condition, to open up and make the rubber very susceptible to corrosive influences.

Fig. 4 shows how the forced amplitude of motion of a vibrating body varies as the ratio of $\frac{\text{forced frequency}}{\text{natural frequency}}$ changes. The "transmissivity" of the mounting is given by:

$$T=\frac{1}{\left(\frac{f}{f_n}\right)^2-1}$$

and the absorption is $1-T$, where f=the forced frequency and f_n=the natural frequency. From the expression for transmissivity it can be seen that the lower the natural frequency of the mounting, the smaller will be the transmitted vibration.

Mountings will be affected by two main considerations. First, the greatest magnitude of the out-of-balance force, and the weight of the engine will determine the quantity of rubber required to keep the stress below its limiting value. In this connection, an acceleration factor of 4½ g in a vertical direction, 1½ g laterally, and 1½ g longitudinally should be allowed for rough riding, cornering, braking, and acceleration. The second condition affecting design is the frequency of the out-of-balance forces. This can vary extensively from idling to full-throttle conditions. It is necessary therefore for the mountings to have a very low natural frequency in torsion, giving low transmissivity when accelerating from slow engine speeds as in traffic. Under these running conditions large fluctuations of torque at a low frequency can be experienced but the vertical out-of-balance forces are not quite so great and mountings of a higher natural frequency in the vertical direction can be used. This is fortunate, since a mounting of low frequency involves large deflections under a given weight, and the amplitude of motion must be limited to a reasonable amount for obvious practical reasons. The natural frequency in the vertical direction is given by $f_n=\frac{188}{\sqrt{\delta}}$ where δ is the static deflection in inches and f_n is the frequency in cycles per second. For further elaboration on the design of engine mountings the reader is referred to "The Theory of Flexible Mountings for Internal Combustion Engines" by C. E. Iliffe.*

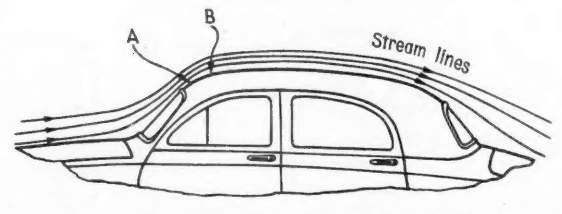


Fig. 5. Front edge of sliding roof at A gives more wind noise than at B

Research and development work need to be done on the subject of isolation of suspension components from the frame. Here, again, the natural frequency of the insulating system must be kept as low as possible, but the conflicting requirements of rigidity and accuracy in the steering system must be met. Manufacturers are at present using proprietary rubber bushes and other similar devices on wishbone and shock absorber joints, and also on rear spring eyes and shackles. In addition, rubber is often interposed between the springs and their axles, and different interleaving materials have been tried. These measures, however, are mostly of a trial-and-error character. Much more definite information is required on methods of insulation and, in particular, on their effect on tyre wear and steering characteristics. Rear shock absorbers need to be well mounted, and their insulation with rubber is a comparatively easy matter.

The vehicle body

Having done all that is possible to isolate the noise-generating parts from the body, it is then necessary to direct attention to the body itself. The first step is to elucidate the actual causes of noise in the body. Wind noise is due of course to the eddying and vibration of air around excrescences which cause a rapid change of the air flow-pattern around a car in motion. There are two main considerations in the design stage in connection with wind noise. The first is to avoid rapid changes of section as far as possible by streamlining window and windscreen arrangements, as well as the overall shape of the vehicle. The second is that such items as an opening for a sliding roof should be in positions where they are least likely to cause turbulence. In Fig. 5 the leading edge of a sliding roof would be much more noisy if located at A on the canopy than at B. At A the streamlines representing the air flow are being made to change direction to conform with the shape of the body, and any slight additional disturbance at this point is liable to cause them to break away and to start eddying. Where the flow is more or less steady along the top of the canopy, as at B, there is less likely to be a break-away. The sliding roof should be flush with the rest of the canopy, and it is important that the fitting of this component should be well executed. There is a need for more research to determine air flow velocities and pressure distribution over motor vehicle bodies. That information would be useful in this connection with ventilation as well as noise problems.

Another cause of noise in bodies is structural unsoundness. A valance or front flitch, for instance, is often designed with no support for the bottom edge. In this condition it lacks stiffness and therefore its natural frequency is low and a large amplitude of vibration is easily excited. Shock absorbers, and mounting brackets in general, are often placed in the middle of a flat panel. Incorrectly and correctly placed exhaust brackets are illustrated in Fig. 6. In the correct method the rubber insulator is square in shape and located in a swage to prevent swinging in a lateral direction, and also to take the offset moment due to the bracket not being in line with the vertical wall.

In thin sheet-metal structures it is a general rule that no load shall be applied to the middle of a panel in a direction normal to the plane of the panel, as this causes the panel to be loaded as a membrane and induces high local bending fatigue stresses at the point of attachment. It is inadvisable to attempt a remedy for an incorrect arrangement, such as in Fig. 6, by inserting a spreader plate at the point of attachment. The only sound remedy would be to

*Proceedings of the Institution of Automobile Engineers, Vol. XXXIV.

place a member between A and B, a channel for instance, to carry the load in bending to the vertical panels which will take it in shear. The best arrangement, of course, is to mount the bracket at A in such a manner that the load is taken directly by the vertical panel in shear. Wherever possible the line of application of the load should be exactly in the plane of the sheet metal in order to avoid local bending due to offsets. Structural unsoundness will lead to panel vibration, rattles and "oil-canning."

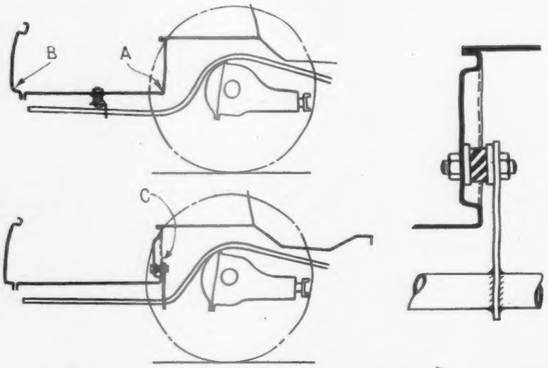


Fig. 6. Incorrect and correct way of mounting exhaust bracket and enlarged view of correct mounting at C

Vibration of panels

One of the more complex causes of noise in motor cars is the natural vibration of the panels. In Fig. 3 it has been shown that in a car the lower frequencies predominate and, therefore, if they could be eliminated a marked improvement would result. It is much easier to reduce the intensity of the higher frequencies with insulating materials, whereas the lower frequencies will not respond favourably to such treatment. The design object is therefore to raise the natural frequency of the panels. Considering a rectangular plate with simply supported edges (not clamped), its natural frequency is given by Timoshenko in "Vibration Problems in Engineering" as :—

$$f_1 = \frac{P}{2\pi}$$

$$\text{where } P = \pi^2 \sqrt{\frac{gD}{\gamma h}} \left(\frac{m^2}{a^2} + \frac{n^2}{b^2}\right)$$

$$\text{and } D = \frac{Eh^3}{12(1-\sigma^2)} = \text{the rigidity of the plate.}$$

f_1 is the fundamental frequency and is obtained when $m=1$ and $n=1$. Other modes are obtained by using different intergers for m and n in turn.

g = gravity

γ = density i.e. $\frac{\gamma h}{g}$ = mass/area

h = thickness

σ = Poissons ratio

E = Youngs modulus.

a and b are the side lengths of the rectangle.

It follows, therefore, that for a square plate

$$f_1 = \frac{\pi}{a^2} \sqrt{\frac{gD}{\gamma h}}$$

Assuming that we wish to limit the natural frequency of panels to a minimum of 200 cycles per second then, from the expression for f_1 above, the dimension of the square panel will be given by

$$a^2 = \frac{\pi}{f_1} \sqrt{\frac{gD}{\gamma h}}$$

substitute for D,

$$a^2 = \frac{\pi}{f_1} \sqrt{\frac{g\,E\,h^2}{12\gamma\,(1-\sigma^2)}}$$

$$a^2 = \frac{\pi}{200} \sqrt{\frac{12 \times 32{\cdot}2 \times 30 \times 10^6 \times 0{\cdot}036^2}{12 \times 0{\cdot}282\,(1-0{\cdot}09)}}$$

$$a = 5{\cdot}86 \text{ in}$$

Thus it can be seen that even to raise the natural frequency to this low value it is necessary to reduce the panel size so that at least one dimension is just under 6 in. In order to visualize what happens when the panel is of rectangular shape, it is necessary to know something of the modes of vibration. Referring to Fig 7, A is the fundamental mode of vibration of a square panel and B, C and D are the higher harmonics. This can be demonstrated by sprinkling sand on a horizontally mounted vibrating panel. If D were bisected by a stiffener at X – Y, it is easy to visualize a fundamental mode for one half of the panel, as shown in E, and the first overtone as in D.

Large curved body-panels vibrate in a much more complex manner. For instance, a door panel such as in Fig. 8 has been found under certain conditions to vibrate so that its points of maximum amplitude are as indicated by the chain-dotted lines A – B, and C – D. Under excitation at different frequencies the pattern or mode of vibration is changed. In the panels of a car, therefore, many different modes of vibration may be excited at closely spaced intervals on the frequency scale. The phenomenon commonly called road roar occurs when most of the panels are vibrating at nearly the same frequency. In other words, they are all in resonance, although their individual modes and harmonics are not necessarily the same. This fact can be proved experimentally in a manner to be described in Part II of this article. Knowing the way in which panels are likely to vibrate, it becomes possible to design for a minimum of road roar. The problem is first to raise the natural frequency of panels above the range likely to give trouble, and then to ensure that they will not all resonate at a common frequency. As far as the internal panels are concerned, e.g. floor and dash, it is a simple matter to divide them up into small panels by means of swages. It is usually found that these panels divide up fairly conveniently into small rectangles of differing natural frequency, which at the same time improves their general stiffness. In the case of the curved outer panels, elaborate bracketing is undesirable but stiffeners are sometimes incorporated for various reasons, such as to support glass channels. If the most likely modes of vibration are visualized, these stiffeners can often be placed so as to interfere with this vibration. A vertical member in the centre of the panel and supporting a window regulator will prevent vibration of a

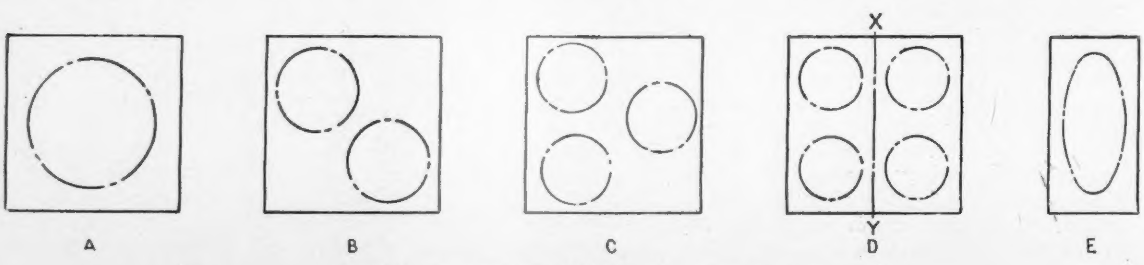


Fig. 7. Some modes of vibration of a square panel

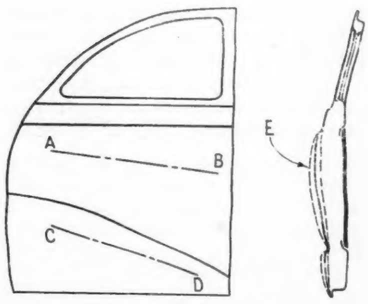


Fig. 8. One of the modes of vibration of a door panel

nature indicated by the dotted lines at E. It would obviously be of little use to incorporate a horizontal stiffener in that particular door, the curvature of which provides sufficient inherent stiffness in that direction.

There are certain fundamental principles which should be applied to swaging. They may be enumerated:

(1) Always make swages straight, as a curved swage tends to pivot about its ends when vibrating

(2) A swage should always take the shortest possible path between supported edges of panels, that is, across the width and not along the length of a panel

(3) Swages should never cross one over the other, since at the point of intersection there is no depth and therefore no stiffness

(4) Stiffness increases very rapidly with increase in depth of swage

Some correct and incorrect methods of swaging are shown in Fig. 9.

Materials applied to panels

Having incorporated the essentials for quietness in the structural design, attention must be given to other means of reducing noise. These may be classified under three headings, (1) insulation; (2) absorption; (3) damping.

The effectiveness of an insulating material is largely dependent on its mass. In view of the fact that it is desirable to keep down vehicle weight in order to improve performance and fuel consumption, it is hardly likely that any completely satisfactory insulation will be found. The 20 S.W.G. sheet steel is far more effective in keeping out noise than almost any of the so-called insulating materials in use today. These materials dissipate sound energy by virtue of the fact that a comparatively large amount of work must be done to maintain their large mass in a state of forced vibration.

Absorption materials, such as felts, work on an entirely different principle. They dissipate the energy of the vibrating air mainly by viscous friction in their porous structure. This class of material will reduce the amount of noise transmitted through it to a small extent, but its chief value is in the prevention of reverberation. An effective demonstration of this is given by using a noise meter in a car with the carpets and under-felt in position and then removed. The difference in the meter readings, at a similar speed on the road in each case, is approximately one decibel where the total noise level is about 85 decibels. On the other hand, the effect of the carpet and felt in reducing noise as perceived by the ear, is far greater than is indicated by the meter. This is explicable on consideration of a single blow on the shell of the car. Such a noise, originating at the floor, will travel to the roof and be reflected back again. It may be reflected backwards and forwards ten times before it loses audibility, thus giving the impression that ten blows have been heard. If the noise in a car on the road is regarded as being a series of blows at a fairly high frequency occurrence, it follows that without carpets or other absorbent material the number of blows will therefore be multiplied. Using the law of addition of decibels, Fig. 2, and allowing for the loss at each successive reflection of a proportion of the energy, it will be seen that reverberation will not increase the decibel level very much. On the other hand, it will increase the noisiness because it increases the frequency of occurrence of the noise.

Nearly all absorbent materials are fairly effective over some parts of the audible frequency range and less effective over other parts. It is therefore necessary to have tests performed on materials in order to determine how useful they are at the frequencies to which they will be subjected. Unfortunately, their efficacy is usually rather poor at low frequencies.

In general, felts are suitable for floors but their absorbent characteristics vary with quality of material. A fine quality fibre-glass bonded with resin is popular in America for roof panels etc. In view of its very light weight it does not tend to become detached. The fine quality does not give rise to handling troubles as does the coarser variety which irritates the skin. This material has the additional advantage that it is not attractive to vermin and does not absorb moisture very readily. Insulgrey and Mutocel, manufactured by L. G. Brown and Co. Ltd. are examples of another suitable material for roofs. This hard felt type of material is in thin sheets, and has enough inherent stiffness to enable it to be supported above listing strips without the use of an adhesive. A light frame can be incorporated if necessary to support the material. It is very wasteful to fit a material such as leather cloth, or fibre-board over an absorbent, as this arrangement reflects sound waves inside the body without their having an opportunity to pass through the absorbent.

There are many proprietary brands of damping material, such as Bittac and Underseal, on the market. Most of these materials are a mixture of bitumen mixed with a substance such as sand, or they are some form of rubber in a suitable solvent. In general it would appear that the most effective materials are those which dry fairly hard. The fundamental principle on which they work is similar to that of a shock absorber in a suspension system. By virtue of their internal frictional resistance, these materials damp vibratory motion of panels. They are most effective when the amplitude of motion is large, such as in a panel in a resonant state of vibration. The selection of material and the question of what thickness to use, can be determined only by experiment, and a suitable method is given in Part II of this article. To be effective, it is absolutely essential to maintain the correct thickness of application in production.

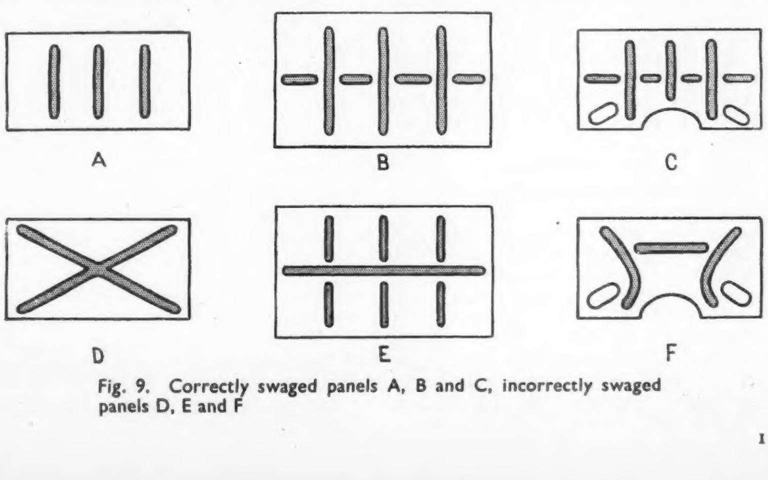


Fig. 9. Correctly swaged panels A, B and C, incorrectly swaged panels D, E and F

Bitumen-impregnated sheets of felt and similar material are also used for damping purposes, and these may be stuck on to panels with a suitable adhesive. Some of these sheets however tend to dry and to curl at the edges. This is a particularly undesirable characteristic in door panels where the curled edges may foul the window actuating mechanism.

It follows from their function that the damping materials must be stuck to the panels. A mistake commonly made is to stick dry felt on to a panel for damping purposes but this material, of course, is only useful as an absorbent. Costs can be unduly raised by incorrect and indiscriminate use of materials.

SYNTHETIC LUBRICANTS

Some Notes on Polyalkylene-Glycols and Their Derivatives

INDUSTRIAL experiences with polyalkylene-glycols and their derivatives are described by C. H. Sweatt and T. W. Langer in an article published in the June, 1951, issue of the American journal *Mechanical Engineering*. These fluids are procurable in water-soluble or substantially insoluble forms, and their range of viscosity overlaps the range of the S.A.E. viscosity numbers 10 to 250, covering crankcase and transmission lubricants. Their qualities include anti-wear action and good load-carrying capacity. They show stability at high temperatures and unusual resistance to sludge and varnish formation.

Internal combustion engine lubrication

The cleanliness of polyglycol fluids, and their ability to dissolve petroleum residues, are applied advantageously in internal combustion engines. Owing to sticking rings, high oil consumption and smoking, certain engines on special purpose vehicles required overhaul after operating five or six months on premium-grade oils. When operating on polyglycol fluids LB-170-X or LB-300-X, the same engines served for periods of two or three years before overhaul. The numbers 170 or 300 represent viscosity in Saybolt universal seconds, the Redwood value being about seven-eighths of this, and X denotes addition of the standard oxidation inhibitor.

LB fluid was employed in one four-cylinder engine when it had reached the overhaul stage but, after several changes of the lubricant which became dirty on dissolving the previously-formed sludge, the engine settled down for a further service of two and a half years. On overhaul, engine, piston and rings proved to be unusually clean and wear was normal. The consumption of lubricant was one quart per 48-hour week.

Gear lubrication

In heavily loaded gears, the anti-wear properties of polyalkylene-glycol lubricants have been strikingly displayed. Bronze ring gears, driven by alloy-steel worms in 75 h.p. heavy-duty speed reducers, suffered severe pitting and spalling at the tooth surfaces when petroleum was used as lubricant, and required replacement after six months of operation. New bronze gears, after a year's operation on LB-1200-X, showed only a trace of tooth-face pitting and were otherwise in excellent condition. Only two quarts of make-up oil per unit were added during the year.

A heavily overloaded speed reducer, having bronze gear-wheel and steel worm, required overhaul, cleaning and replacements at three-monthly intervals. Even so, failures occurred between overhauls. High wear and severe carbonization occurred, and excessive sludge formed while various petroleum products were employed as lubricants. With LB-1200-X, however, practically no sludge formed and there was no evidence of excessive wear or spalling after fifteen months of satisfactory operation. Bearings ran at lower temperatures.

> **THE PARIS SALON**
>
> *A special detailed report of the 1951 Paris Salon will appear in the issue for December, published December 13th. It will constitute a survey of the more interesting designs, with an indication of bodywork trends.*
>
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A charge of LB-300-X, in the transmission and hypoid rear axle of a laboratory test car, was changed in the course of 135,000 miles only three times in the rear axle and not at all in the transmission. The performance was entirely satisfactory, and gears and bearings were in excellent condition after the test.

Unusually high film strength and load-carrying qualities, under certain conditions, may be demonstrated in the bench-type lubricant testing equipment. A modified test, in which the Timken bearing races are rotated while being held against each other, simulates the rolling-and-rubbing action of a hypoid gear. At certain speeds, loads and rubbing ratios, scoring occurs almost immediately with uncompounded mineral oils, though some petroleum products containing extreme-pressure additives complete a 15-hour test at 470 r.p.m. with little wear, under a load of 135 lb. and with a rubbing ratio of 3·4 : 1. Polyglycol lubricant LB-140-X plus 2-105B extreme-pressure additive returned the lowest weight loss at the bearing races, however, and even some uncompounded LB-type lubricants, having viscosities not less than 300 seconds, were able to complete the test, the wear being comparatively little.

Metal working applications

When LB and 50-HB polyalkylene-glycol type fluids were substituted for conventional oils in drawing operations on brass, sheet iron and nickel-plated steel shells, and in blanking and pressing annealed spring steel, about three times as many parts could be produced before tool drawing, die cleaning or die reconditioning were required. Before brazing aluminium parts, the lubricant used in a previous process had to be burned off, and distortion resulted. A 50-HB synthetic lubricant could be removed by water-washing. By the use of polyglycol fluids, superior finish was obtained in the lapping of stainless steel threads with glass or diamond dust.

Hydraulic fluids

Synthetic products are being employed as brake fluids and in machine tools, presses and outdoor instruments. The relatively small variation of viscosity with temperature render them especially suitable for use in cold weather, and DLB-50-B, for example, may be used under pressures up to 150,000 lb per sq in. In pumps in which boundary lubricating conditions prevail, there is likely to be reduction in wear, and leakage at packings is greatly reduced.

Polyalkylene-glycol thickening agents are combined with water and other substances to form the non-inflammable Hydrolube compositions used in U.S. Navy aircraft and hydraulic applications in industry. (1978)

Rubery Owen of Darlaston, South Staffs have issued a booklet entitled "An Industrial Commonwealth". It describes the work of almost every section of the firm and of its subsidiary companies and its 104 pages include many excellent photographic illustrations. A comprehensive list of products and services, which includes such items as bricks, machine tools and chains as well as automobile parts, shows the very wide range of work undertaken by this organization.

GAS CARBURISING

An Economical Method of Case Hardening

SOLID, liquid and gaseous media are all used for carburising steel. Solid carburisers are still the most widely used, but in recent years gas carburising has been used in an ever increasing degree. This is not surprising since this method has undoubted advantages, particularly where large components requiring relatively deep cases have to be treated, and precise control of case is essential. Special equipment is, of course, necessary, and these notes deal with equipment developed by Wild-Barfield Electric Furnaces Ltd., Elecfurn Works, Watford By-pass, Watford, Herts.

Where "natural gas" is not available, carburising atmospheres are normally of the high hydro-carbon type suitably diluted. Typical examples are butane or propane diluted with charcoal burner gas, raw town's gas or burnt town's gas. Such hydro-carbons possess the characteristic of "cracking" on the surface of the steel at temperature, and give rise to carbon deposition which takes the form of a hard adherent scale. Inevitably, the results of such deposition are the slowing of the carburising rate and the bad fouling of retorts. In such circumstances it is not possible to carry out the carburising under precisely controlled conditions, since the "stopping off" effect may vary, and unless precise control is available the full benefits of gas carburising cannot be obtained.

Wild-Barfield equipment for gas carburising includes means for preparing raw town's gas in such a manner that the "stopping off" effect is eliminated. Town's gas direct from the mains is fed to a compressor and thence through a flow meter to a special catalyst, through an air cooler to driers and then direct to the carburising retorts. The gas from the catalyst exit passes through the air cooler to silica gel driers. Only one drier is used at a time so that the plant can be operated continuously. Regeneration of the silica gel is simply effected by a low temperature treatment in a small forced air circulation furnace. After passage through the drier, the gas goes via an inferential meter to the carburising retorts. The action of the catalyst employed in the Wild-Barfield process may be seen from Table I.

It will be seen that the methane content is only slightly reduced and there is a considerable increase in the amount of carbon monoxide. Town's gas supplies vary according to the method of production, but invariably if the methane content is low, the carbon monoxide content is correspondingly high.

TABLE I

ANALYSES OF TOWN'S GAS BEFORE AND AFTER PASSING THROUGH THE CATALYST

	Raw town's gas per cent	Prepared town's gas per cent
Carbon dioxide	3·3	—
Oxygen	0·6	—
Unsaturated hydro-carbons	2·1	0·6
Carbon monoxide	16·4	20·2
Methane	24·0	23·6
Hydrogen	49·5	51·2
Nitrogen (balance)	4·1	4·4

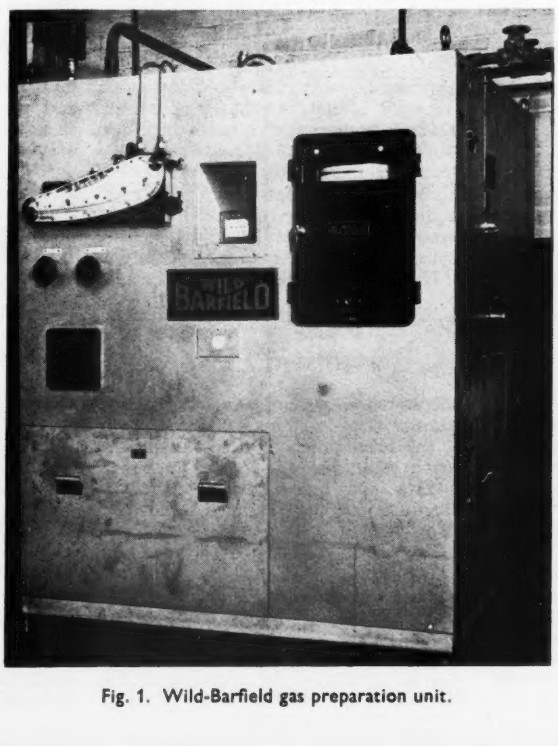


Fig. 1. Wild-Barfield gas preparation unit.

In operation the retort is first purged and then lowered into a furnace controlled at the specified carburising temperature. The gas flow, as read on the flow meter is then adjusted, and no further attention is required until the "active" carburising period is finished. At the end of this period the compressor is stopped and the outlet valve of the retort is closed. While the charge is completing the diffusion period, the preparation unit can be used to purge other retorts.

Catalyst and driers are regenerated after a certain volume of gas has passed through them for purification. This volume is read off the inferential meter. The frequency of regeneration is dependent upon the volume of gas passed; but normally it does not exceed once per week even when the plant is in use for 24 hours per day. Regeneration of the catalyst can conveniently be carried out while a charge is being diffused and the preparation unit is not required. As two driers are provided, continuous use of the plant is possible.

TABLE II

CASE DEPTH AT MAXIMUM CARBURISING RATE

Material	Temperature Deg C	Deg F	Time hours	Case depth in
Mild steel (S/4)	925	1697	1	0·025
	925	1697	3	0·043
	950	1742	1	0·032
	950	1742	3	0·055
S.82	925	1697	5	0·050
2 per cent NiMo C.H. and 3 per cent Ni	925	1697	1	0·023
	925	1697	3	0·039
	950	1742	1	0·029
	950	1742	3	0·040

Results

The figures in Table II give examples of the case depths obtained by carburising at the maximum rate. Extensions of these results can easily be calculated if use is made of the fact that, given available carbon at a rate always greater than the rate of diffusion into the particular steel at the specified temperature, the case depth is proportional to the square root of the carburising time. That is, to double the case, the total time must be increased four times.

Practical advantages

The equipment is fed with raw town's gas direct from the mains and prepares it for carburising without any additions such as butane and

Fig. 2. A typical gas carburising installation comprising three Wild-Barfield forced air circulation furnaces. quench tank and gas preparation unit.

propane. All the deleterious constituents of the town's gas are removed. Extremely precise control of the depth and type of case is afforded. The surface of a component can therefore be ground without appreciable decrease in carbon content and consequent loss of hardness. Furthermore, the control of the carbon gradient ensures that too high a carbon content at the surface is avoided. This removes any danger that the hardened surface will crack during a grinding operation. In addition, a greater proportion of the total case is hardenable and the hardness transition from case to core is more gradual.

With prepared town's gas the time for treatment can be reduced. It does not leave a hard carbon deposit upon the work with a consequent "stopping off" effect. Any slight deposit from prepared town's gas is in the form of pulvurent graphite which has no "stopping off" effect. Therefore there is no need to obtain any balance of mixture to minimize deleterious sooting, and carburising can proceed at the correct temperature for the steel under treatment. Furthermore, the precise control over diffusion allows the "active" carburising to be carried out at the maximum theoretical rate. These factors give faster production with better quality.

As town's gas is used without any additions, the cost of the carburising atmosphere is appreciably lessened. In addition, since the greater part of the cost of carburising is attributable to the heating time, the shorter time of treatment materially reduces the overall cost. Furthermore, in comparison with box carburising, savings are effected in the labour costs involved in packing, unpacking and handling boxes, and in the costs of carburising material, its storage and removal. There is also simplicity of operation. The gas preparation unit has very few controls. Once the relevant valve settings have been made, the complete process may proceed without supervision apart from stopping a compressor and closing the retort valves when the active period ends and the diffusion begins. For a given output a gas carburising installation requires much less space than would be needed for furnaces for pack carburising. A typical installation is shown in Fig. 2.

BRITISH STANDARDS

*B.S.*1083:1951. *Precision hexagon bolts, screws, nuts (B.S.W. and B.S.F. threads) and plain washers (price* 4*s.* 0*d.)*

The present edition of B.S.1083 confirms as a regular British Standard the war emergency Standard which was issued in 1942 at the request of the Ministry of Supply in place of B.S.191 and B.S.193. After the end of the war period, the question of the most suitable post-war British Standard for bright bolts and nuts was discussed, and after consultation with a wide field of user interests, it was agreed that B.S.1083 should be re-issued.

In preparing the new edition the opportunity has been taken to introduce certain amendments. The basic dimensions remain unchanged, and the amendments are concerned principally with a clarification of certain of the former requirements and the inclusion of additional requirements to improve the general quality of the bolts and nuts. The term "precision bolts" is now used to designate the type of bolts, together with the corresponding screws and nuts, which are manufactured to prescribed mechanical properties and to tolerances required for use in engineering work where a good standard of dimensional accuracy and performance is required.

The Standard relates to ferrous and non-ferrous bolts, screws and nuts, and to split cotter pins and washers for use with them, and in the case of steel refers to the appropriate qualities in various grades up to 85 tons tensile as specified in B.S.970. It gives the detailed dimensions for bolts, screws, nuts, lock nuts, castle nuts, split pins, and washers for all regular nominal sizes from ¼ in. to 2 in. Stock sizes of steel bolts and screws and a code of part numbers are given in the appendices.

Copies of this Standard may be obtained from the British Standards Institution, Sales Department, 24 Victoria Street, London, S.W.1.

Reinforced Plastics

In *The Engineers' Digest* of May 1951, it is stated that glass fibre polyester laminates are a very suitable material where high mechanical strength and low weight are required. In the past, the use of such laminates, mouldings and reinforced sheets was limited by the fact that their mechanical strength is radically impaired on exposure to water or high humidity. In fact, these materials lose as much as 50 per cent of their flexural strength on immersion in water or exposure to high atmospheric humidity for any extended period. According to a recent announcement, this difficulty has now been overcome by the development of a new manufacturing technique so that the improved material may serve as an excellent substitute for steel and light alloys in many cases.

The new technique is based on recognition of the fact that, in the original method, the bond between fibres and the plastic was not strong enough to withstand the action of water. It was found that this weakness can be eliminated by applying to the glass fibre a size consisting of vinyl chlorosilane, followed by a wash with water. It appears that in this manner the silicone atoms in the vinyl chlorosilane molecules become tied to the glass, while the vinyl part of the size will react and participate in the polymerization of the resin, so that an actual bond is formed between the glass and the plastic. Particularly promising applications of the new material are claimed to be automobile tops, electric stoves, laundry machines, refrigerator casings, filing cabinets, etc.; these are in addition to a great many military uses. (1969)

CURRENT PATENTS

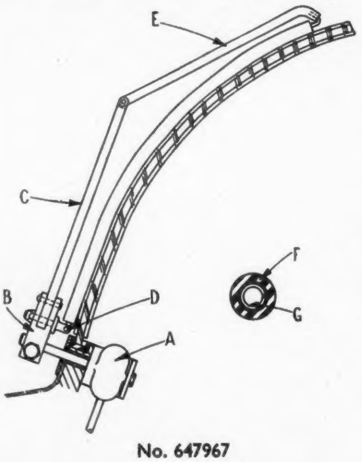
A Comprehensive Review of Recent Automobile Specifications

Windscreen wiper

THIS wiper is self-adjusting to maintain the blade in wiping contact with a convexly curved surface. Attached to the spindle of a conventional motor A, a member B carries a rigid arm C and an eye-bolt D. To the end of arm C is pivoted a movable arm E which is free to swing about its pivot only in the plane containing the arm C and the motor spindle.

The wiper blade is constituted by a rubber sleeve F containing a helical spring G having at each end a projecting hook. One hook is engaged in the eye-bolt and the other in the extremity of the arm E. The relation between the dimensions of the wiper and the shape of the screen must be such that when mounted in position the wiper blade is under tension. As a consequence the blade tends to resume its free state, bears closely on the screen and, by reason of its resilience, conforms to the shape of the screen.

As a modification the wiper blade may comprise a rubber member fitted with a metal attachment at each end. The specification also describes a wiper arranged for reciprocating movement over a curved



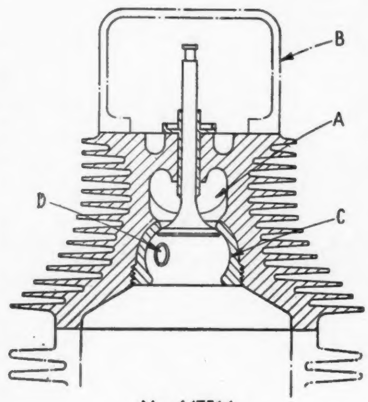
No. 647967

screen. In this the actuating arm carries a plate to which is pivotally mounted a pair of arms supporting the resilient blade. *Patent No.* 647967. *Dunlop Rubber Co. Ltd. and H. W. Trevaskis.*

Cylinder head for air-cooled two-stroke diesel engine

THIS finned cylinder head is constructed of a material of high thermal conductivity and fitted with a liner of more heat-resistant material to shield it from the combustion flame. The body of the head is in a copper alloy, such as aluminium bronze, and is proportioned to ensure a rapid transfer of heat from the more thermally stressed localities. In its upper portion is embodied the exhaust port A carrying the valve guide while the flat top surface forms a seating for the rocker box B.

A substantially hemisphero-cylindrical combustion chamber is furnished with a high-expansion steel or cast iron liner C



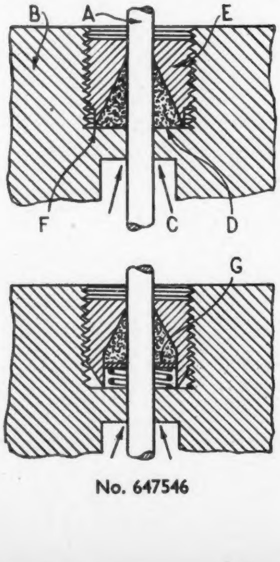
No. 647514

which may be cast in or, as shown, screwed in with an interference fit. The exhaust valve is seated in the hard metal liner, which is also apertured at D to admit the fuel injector. *Patent No.* 647514. *E. E. Pobjoy (D. R. Pobjoy).*

High-pressure stuffing box

WHERE high pressures, of the order of 60 atm for example, have to be supported, the conventional stuffing box in which the packing is clamped between two coned shoulders will not long maintain fluidtightness unless the loading is increased to compensate for wear. Automatic loading devices are liable to be somewhat complicated and expensive and the object of this invention is to provide a simple construction that is capable of sustaining high pressures for prolonged periods.

In the diagrams, shaft A is movable axially to and fro through the body B and is subjected to pressure acting in the direction of arrows C. The body is provided with a threaded recess concentric with the shaft and terminating in a flat base. D. Into this recess is screwed the gland element E having a frusto-conical seating into which is forced a tapered



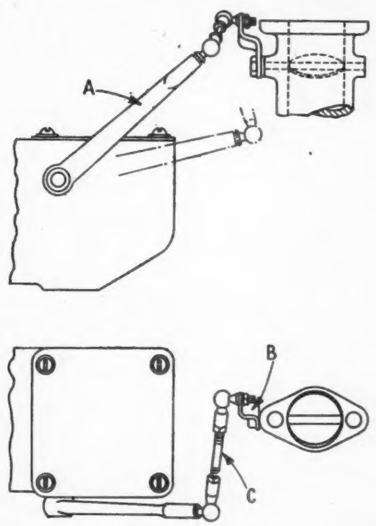
No. 647546

packing ring. The lower face of element E is machined to a sharp-edged abutment F which embeds itself in face D to form a fluidtight joint. The fluid tends to force the packing ring into close contact with the shaft.

In an alternative construction, the frusto-conical seating has an additional cylindrical portion and the packing ring is correspondingly shaped. The packing ring is subjected to constraint by a compression spring acting on a thrust washer G. *Patent No.* 647546. *Motosacoche S.A. (Switzerland).*

Governor linkage

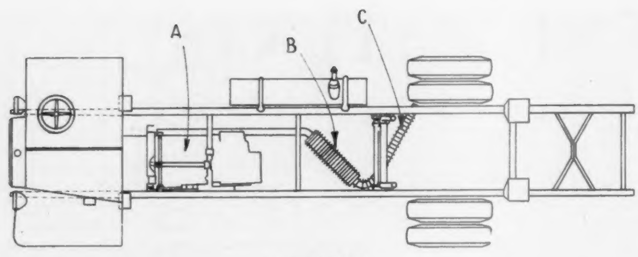
BY means of a linkage arrangement taking virtually the form of a three-dimensional toggle, a change of torque more uniform than is usual is obtained in response to control by a centrifugal governor. The axis of the governor spindle, on which is mounted lever A, lies in a vertical plane substantially at



No. 647991

right angles to a vertical plane containing the axis of the throttle valve spindle, fitted with a crank arm B. A link C, ball-jointed at each end, couples governor and throttle levers and the arrangement is such that at all times lever A is offset from a plane containing the axis of the governor spindle and the ball joint on throttle arm B.

With the throttle in the closed position, as shown, a high velocity ratio exists between the linked ends of the two levers, but the ratio is lowered as lever A moves the throttle towards the fully open position. In other words, a constant rate of movement of the governor lever produces an increasing rate of opening movement of the throttle. Another advantage of the arrangement is that the mechanical advantage is highest when the throttle is nearly closed and offers most resistance to movement as a consequence of the enhanced depression in the inlet manifold. *Patent No.* 647991. *Wolseley Motors Ltd.*

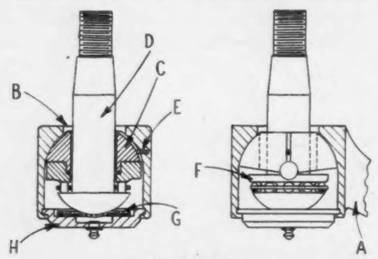


No. 646679

Ball joint for independent front suspensions

FOR the steering linkage of independent front wheel suspensions of the so-called "knee-action" type a ball stud must be rotatable to permit free steering while tilting in one plane to accommodate the knee action and change of camber angle and in another plane to allow change of castor angle. The joint in this invention accommodates the three movements on separate, independent bearings.

In the end of wheel arm A is formed an inverted cup-shaped chamber, the segmental spherical surface of which converges to a circular opening B. A bearing member C, seating on the spherical surface, has an elongated central aperture to receive the shank of stud D which it engages with its minor dimension. Thus the stud can tilt freely in one plane but carries the bearing C with it in a plane at right angles, for castor variation. Direction of movement of the bearing is determined by a pin E engaging a groove in the bearing surface.



No. 648413

Diametrically opposite, semicircular recesses in the base of the bearing and lying on the plane of movement, receive the trunnions of a ring F. This ring serves as the upper race of an anti-friction bearing assembly, in this instance caged rollers, while the lower race is furnished by the shoulder of the part-spherical head of the stud. For knee-action and camber variation the stud tilts about the trunnions, and under all conditions is freely rotatable about its own axis.

The assembly is completed by a spring seat G engaging the stud head and constrained by a spring washer supported on the closure plate H. *Patent No.* 648413. *Thompson Products, Inc.* (*U.S.A.*).

Dissipating electrostatic charges

UNDER certain conditions a vehicle may accumulate an electrostatic charge which may be at a high potential to the ground potential. The rate of leak depends upon the resistance of the vehicle chassis to the ground and any person establishing contact with the vehicle while standing on the ground is liable to receive an electric shock. Proposals to reduce the charge by means of a trailing conductor are not acceptable from the point of view of wear and noise. Tyres have been constructed with electrically conductive walls and treads but these are difficult to produce and their value is liable to vary with increasing wear.

The invention proposes to use the water content of the exhaust gases to establish an electrically conductive path between the vehicle body and the ground. The exhaust gases may be directed, either continuously or intermittently, on to the ground or on to the wheel rim and tyre.

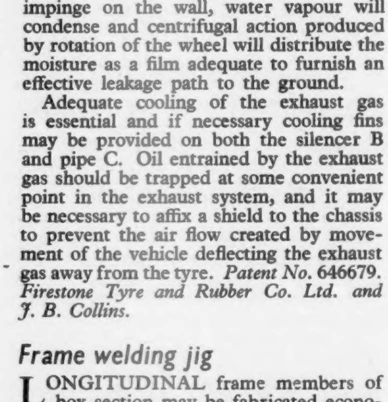
The illustration shows a public service vehicle equipped in this manner. Exhaust gas from engine A is passed through the silencer B and then directed by way of pipe C to impinge partly on the rim flange and partly on the tyre wall so that a continuous film of water is formed between the rim and the ground. Droplets of water will impinge on the wall, water vapour will condense and centrifugal action produced by rotation of the wheel will distribute the moisture as a film adequate to furnish an effective leakage path to the ground.

Adequate cooling of the exhaust gas is essential and if necessary cooling fins may be provided on both the silencer B and pipe C. Oil entrained by the exhaust gas should be trapped at some convenient point in the exhaust system, and it may be necessary to affix a shield to the chassis to prevent the air flow created by movement of the vehicle deflecting the exhaust gas away from the tyre. *Patent No.* 646679. *Firestone Tyre and Rubber Co. Ltd. and J. B. Collins.*

Frame welding jig

LONGITUDINAL frame members of box section may be fabricated economically by welding together four suitably shaped strips, or a pressed channel and a strip, or two channel elements. In all cases difficulty may arise in maintaining two generally-parallel elements in accurately spaced relationship, while another element is being affixed. It is proposed in this invention to support the work between spaced lines of electromagnets.

In the jig shown the magnets A of each line are arranged end-to-end, with their pole pieces B projecting inwardly at intervals of about 4 in, on side members C and D pivotally mounted on the base structure. Member D is held in operative position by a quick-release clamp E and carries the pivoted beam F of the top clamp. Using current of low voltage for reasons of safety, say 18 volt, a suitable magnet will exert a pull of about 35 lb.

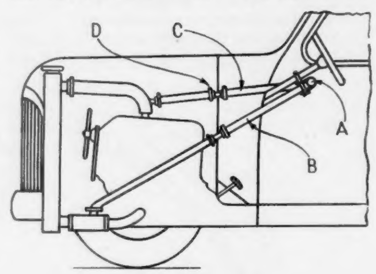


No. 648173

For a four-strip frame member, two strips are placed on edge on wooden blocks G, the magnets are energized to position them, and the uniting strip is clamped on the upper edges and tack-welded. After removal from the jig the three strips are seam-welded to form a channel section. It is then inverted, mounted in another jig of the same type, and the fourth or closure strip is tacked on preparatory to final seam-welding. *Patent No.* 648173. *Rover Co. Ltd.*

Thermo-syphon heater

PARTICULARLY intended for small vehicles fitted with an engine having a thermo-syphon cooling system, this interior warming arrangement can be installed without extensive structural alteration. It comprises a tube of substantial length, closed at each end and provided with means for mounting in a horizontal position on the steering column at a level above that of the top of the engine but slightly below the top of the radiator. Flow and return pipes connect opposite ends of the tube with the top and bottom hose connections of the engine cooling system. It is claimed that after a few minutes running the tube is raised to a temperature sufficient to give out considerable heat and, with continued running, maintain the interior temperature of the vehicle at a comfortable level.



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Heater tube A is of steel and is closed at each end by a flanged cap. Adjacent to one end is a short tubular connector adapted to receive a flexible hose B, while from the opposite end is a similar connector for hose C. The heater tube may be attached to the steering column by means of a pair of U-bolts to permit angular and axial adjustment. A small bleed hole, closed by a screw, is provided in the upper surface of the tube to permit the escape of air when the engine system is refilled after draining.

The end of each flexible hose is taken to an appropriate point in the bulkhead where it is attached to a rigid metal connector D and from the opposite side of the bulkhead a further hose establishes connection with the engine system. It is essential that the holes in the bulkhead are sited so that the flow and return pipes rise in an approximate straight line from the engine to the heater.

Experiment has shown that the heater functions most efficiently if the top of the tube is about 1½ in below the level of the top of the radiator. A cock for regulation of the flow may be fitted in either the flow or return pipe. *Patent No.* 648496. *E. Able.*